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# THE CUMMINS ADVANCED TURBOCOMPOUND DIESEL ENGINE EVALUATION

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## FOREWORD

This technical report covers all the engine test activity necessary to fulfill the scope of work requirements of Contract DE-AC02-78CS54936 between the Department of Energy and Cummins Engine Company.

The government program management was conducted by the Office of Vehicle and Engine R&D. This organization is within the auspices of the Assistant Secretary for Conservation and Renewable Energy of the Department of Energy. Program officers of the Department of Energy were Mr. S. B. Kramer, Mr. A. A. Chesnes, and Mr. E. W. Gregory, II. Under the terms of a DOE/NASA interagency agreement, the NASA-Lewis Research Center of Cleveland, Ohio, served for DOE as the technical project managers for this project. James C. Wood served as the technical representative of the NASA-Lewis organization.

The requirements of NASA Policy Directive NPD 2220.4 (September 4, 1970) regarding the use of SI Units have been waived in accordance with the provisions of paragraph 5d of that Directive by the Director of Lewis Research Center.

The Cummins technical director of this program was Mr. Roy Kamo. The authors would like to acknowledge the valuable contribution in the performance of the program by the following people: M. C. Brands, M. Cooper, C. J. Rhoades, J. Cox, J. M. Mulloy, and H. G. Weber.

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## SUMMARY

The primary objective of the advanced turbocompound diesel engine program was to improve the tank mileage by 5% over the 1980 vehicle test (interim) turbocompound diesel engines. The technical approaches used to develop the advanced turbocompound engine were:

- I. Increase the available exhaust gas energy to the turbines with a more efficient exhaust manifold and by insulating the exhaust system components.
- II. Improve the fuel injection characteristics by providing higher injection pressures and shorter injection durations.
- III. Improve the turbocompound system by optimizing the power turbine speed for maximum turbine efficiency and by reducing the turbine shaft bearing mechanical losses.
- IV. Lower the intake manifold temperature from 140°F to 110°F to reduce nitric oxide emissions, and increase engine thermal efficiency.
- V. Improve the compressor efficiency 1-2% by reducing the operating clearances with an abradable shroud.

The combined effect of these improvements resulted in a rated power BSFC of .310 lb/bhp-hr with a minimum BSFC of .298 lb/bhp-hr while meeting the California 6 gram combined ( $\text{BSNO}_x + \text{BSHC}$ ) gaseous emission level.

To quantify the tank mileage improvements of the advanced turbocompound engine its performance map was used as input to Cummins' Vehicle Mission Simulation (VMS) program to predict the tank mileage over the Cummins' Pilot Center fuel economy route for comparison to the interim engines. In the course of the vehicle testing program completed in 1980, it was proven that an excellent correlation exists between VMS predicted tank mileages and actual vehicle test results. The VMS calculation predicted a tank mileage of 5.75 mpg for the advanced turbocompound engine while the interim turbocompound engine prediction was 5.40 mpg. Thus, a predicted tank mileage improvement of 6.5% was achieved with the advanced turbocompound diesel engine.

In summation, the advanced turbocompound diesel engine program met and exceeded all tank mileage goals, further enhancing the potential fuel consumption savings of the turbocompound diesel engine.

## INTRODUCTION

A program has been underway at Cummins Engine Company, Inc. since 1972 to develop the turbocompound diesel engine. This engine is a hybrid diesel reciprocator which is augmented in power by a low pressure power turbine. The turbine power generated by means of exhaust gas expansion is transferred to the drive train by mechanically gearing the power turbine to the rear of the crankshaft at a fixed speed ratio. A fluid coupling is utilized to separate the crankshaft torsional vibrations from the high speed gearing and turbine shaft.

The laboratory engineering development of the turbocompound engine reached a mature stage following the evolution of three power turbine and gear train designs. The next logical evaluation involved vehicle performance testing. Thus, Cummins Engine Company entered into a contract with the Department of Energy in 1978 which called for a comprehensive vehicle test evaluation to ascertain the viability of the turbocompound engine for trucks and buses of the 1980's.

As part of this effort, two 450 BHP NH turbocompound diesel engines were assembled and dynamometer tested. Both engines met the California 6 gram (BSNO + BSHC) combined gaseous emissions limit and achieved a minimum fuel<sup>x</sup> consumption of .313 lb/bhp-hr and a value at rated power of .323 lb/bhp-hr. These engines were then installed in Class VIII (73,000 GVW) heavy-duty trucks to determine their fuel consumption potential and performance characteristics. One turbocompound engine-powered vehicle was evaluated at the Cummins Pilot Center facility where detailed engine-transmission-vehicle tests were conducted in a controlled environment. The other engine was placed in commercial service operating between Florida and California for 50,000 miles. The results of these tests are reported in the NASA Technical report CR-159840 (Ref. 1). The most salient finding was that the turbocompounded engines in both locations showed a fuel consumption reduction of 15-16% over the production NTC-400 horsepower reference engine.

During these tests, a number of component modifications were incorporated in the turbocompound engine which resulted in fuel consumption reductions exceeding the expected benefit from turbocompounding alone. Through previous laboratory testing, it had been established that a benefit of 6% reduction in fuel consumption over an equivalent turbocharged and aftercooled NH engine was achieved with turbocompounding along the engine's torque curve. As the load is reduced, the gain reduces in value as the available exhaust energy decreases. Using these test results, the incremental fuel consumption improvement due to the turbocompounding alone was 4.2% to 5.3% for the interim turbocompounded engine, depending upon the terrain or mission load factor.

The vehicle testing activity described above was conducted with the interim turbocompound diesel. The interim nomenclature is primarily a distinction regarding the development status of the engine at the time of the vehicle test activity. That is, the

vehicle test engines had known aerodynamic deficiencies and utilized modified production components to turbocharge the engine. Improvements could be realized by increasing the available energy of the exhaust gas at the turbine inlet, by incorporating current aerodynamic design practices into improving the turbomachinery efficiencies, and through refined engine system optimization. The combined effects of these improvements were expected to increase the tank mileage by 5% over the interim turbocompound diesel engines. Therefore, in September of 1980, the Department of Energy extended the contract with Cummins to continue the development of an advanced turbocompound diesel engine. The primary objective of this program was to improve the tank mileage of the advanced turbocompound diesel engine by 5% over the interim turbocompound diesel engine. A five-phase program was established to achieve these goals and enhance the fuel conservation potential of the turbocompound diesel:

- Task I      Baseline Performance Mapping
- Task II     Engine Performance Testing and Upgrading
- Task III    Advanced Engine Preparation
- Task IV     Advanced Engine Dynamometer Testing
- Task V      VMS Analysis

Task I included engine removal from the Pilot Center test vehicle and installation in a dynamometer test cell to establish baseline data repeatability. Steady-state performance mapping consisting of a matrix of engine speed and load conditions was completed along with measurement of gaseous emissions.

Task II included an improvement in the fuel injection system along with flow path design changes to the exhaust manifold to reduce the pumping losses and minimize the exhaust gas mixing losses.

Task III entailed the implementation of existing design practices typically employed in gas turbine power plants into the turbocompound engine turbomachinery, engine system optimization, and insulation of the exhaust system components.

Task IV was the dynamometer testing of the advanced turbocompound engine developed in Task III. This included performance mapping to assess the performance achieved against the interim engine performance at equivalent gaseous emission levels.

Task V utilized the performance map generated in Task IV as input to Cummins' Vehicle Mission Simulation (VMS) computer program. VMS is an analytical model which predicts the tank mileage for the advanced turbocompound engine.

These VMS results were, in turn, compared to the reference tank mileage predictions of the interim turbocompound vehicle test engines.



## 1.0 Turbocompound Engine Description

The turbocompound engine was developed from the Cummins NH engine. The bore and stroke of the NH engine are 5.5 and 6.0 inches, respectively. The NH is four cycle inline-six cylinder of 855 cubic inch displacement. Fuel is supplied to the engine by the Cummins PT® high pressure injection system. The turbocompound engine was turbocharged, aftercooled, and conventionally cooled.

A number of component modifications have been made to improve the engine system performance under turbocompounding conditions. These include design changes of the camshaft, valves, cylinder head exhaust ports, exhaust manifold, and turbocharger. Primarily, these modifications were initiated to reduce the blowdown energy losses during the exhaust phase of the cycle and to improve the transmission efficiency of the exhaust gas from the cylinder to the first stage turbine.

The turbocompound system consists of three separate modules. The modular concept was selected to provide for ease of assembly and maintenance. The first module consists of a radial inflow low pressure power turbine to recover the exhaust gas energy and its bearing cartridge. The second module consists of the high speed gearbox using involute spur gearing to achieve part of the necessary speed reduction from the power turbine to the crankshaft. Lubrication is provided by the engine oil system and directed by internal oil drillings. The third module is the low speed gearbox which completes the speed reduction required. A fluid coupling is an integral part of this module which performs the function of separating the high speed gearing from the crankshaft torsional vibrations. The flywheel housing is an S.A.E. No. 1 housing constructed of cast iron to support the weight of the gear train. The overall increase in engine length is one inch and the entire system is designed such that it may be installed in most high horsepower engine applications. The design provides for 50% overspeed capability and 100% overspeed burst containment. A schematic of the Cummins turbocompound diesel engine is shown in Figure 1, while the assembled engine is shown in Figure 2.

The turbocompound engine was rated 450 brake horsepower at 1900 rpm engine speed with 15% torque rise to 1440 lb-ft at 1300 rpm. A lower operating speed rating was selected to take advantage of the low speed torque characteristics of a turbocompound engine. The increase in power rating at a lower engine speed was achieved without increasing the thermal or structural loading of the reciprocator.

## 2.0 Baseline Testing: Task I

The interim turbocompound engine was removed from the Cummins Pilot Center test vehicle in April of 1980. This engine and the associated turbocompound gear train were disassembled and examined for any undue or unusual wear. Inspection revealed that the high speed gear housing bushing had seized on the fluid coupling shaft.

Figure 1

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CUMMINS NH  
TURBOCOMPOUND  
DIESEL ENGINE

A hybrid diesel-turbine system  
in which piston power is  
supplemented by turbine power  
recovered from the exhaust  
gas.

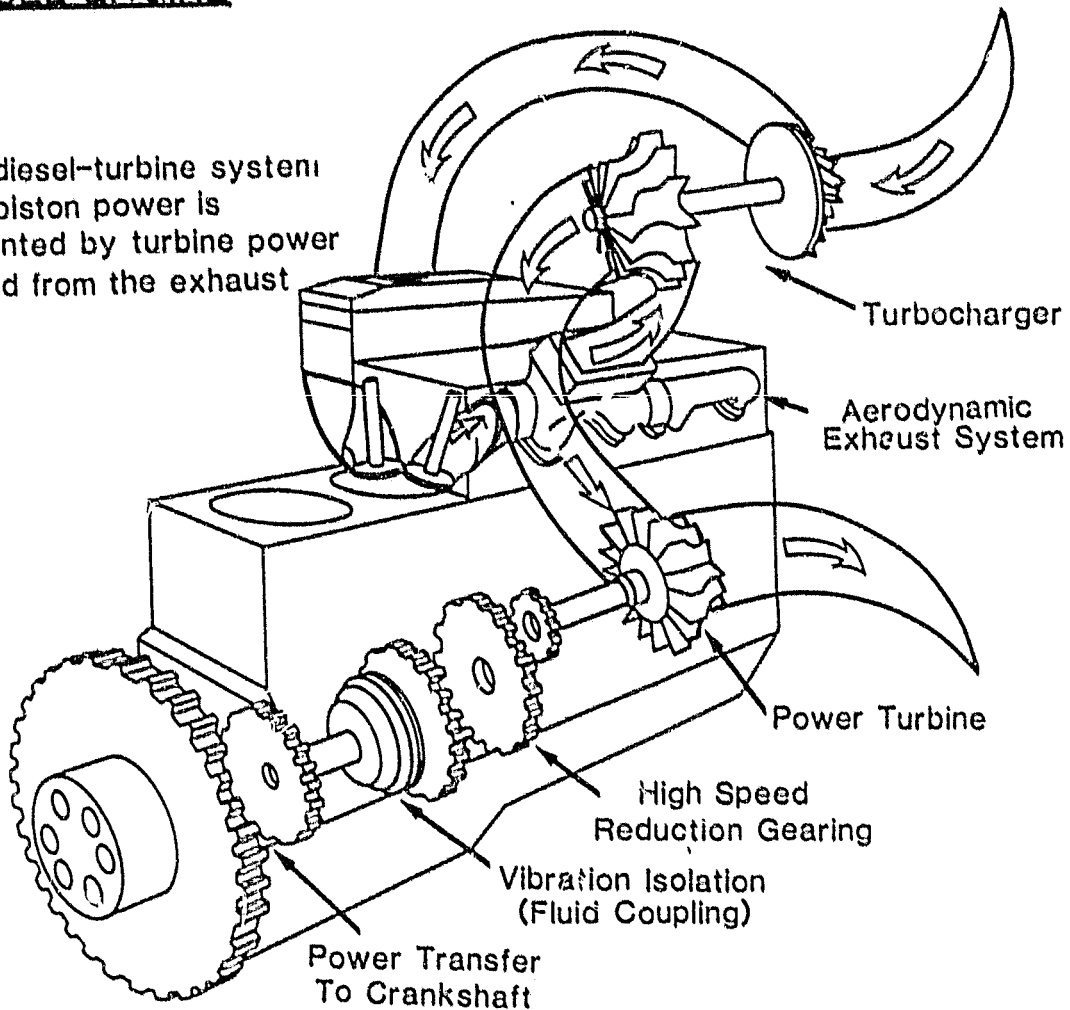
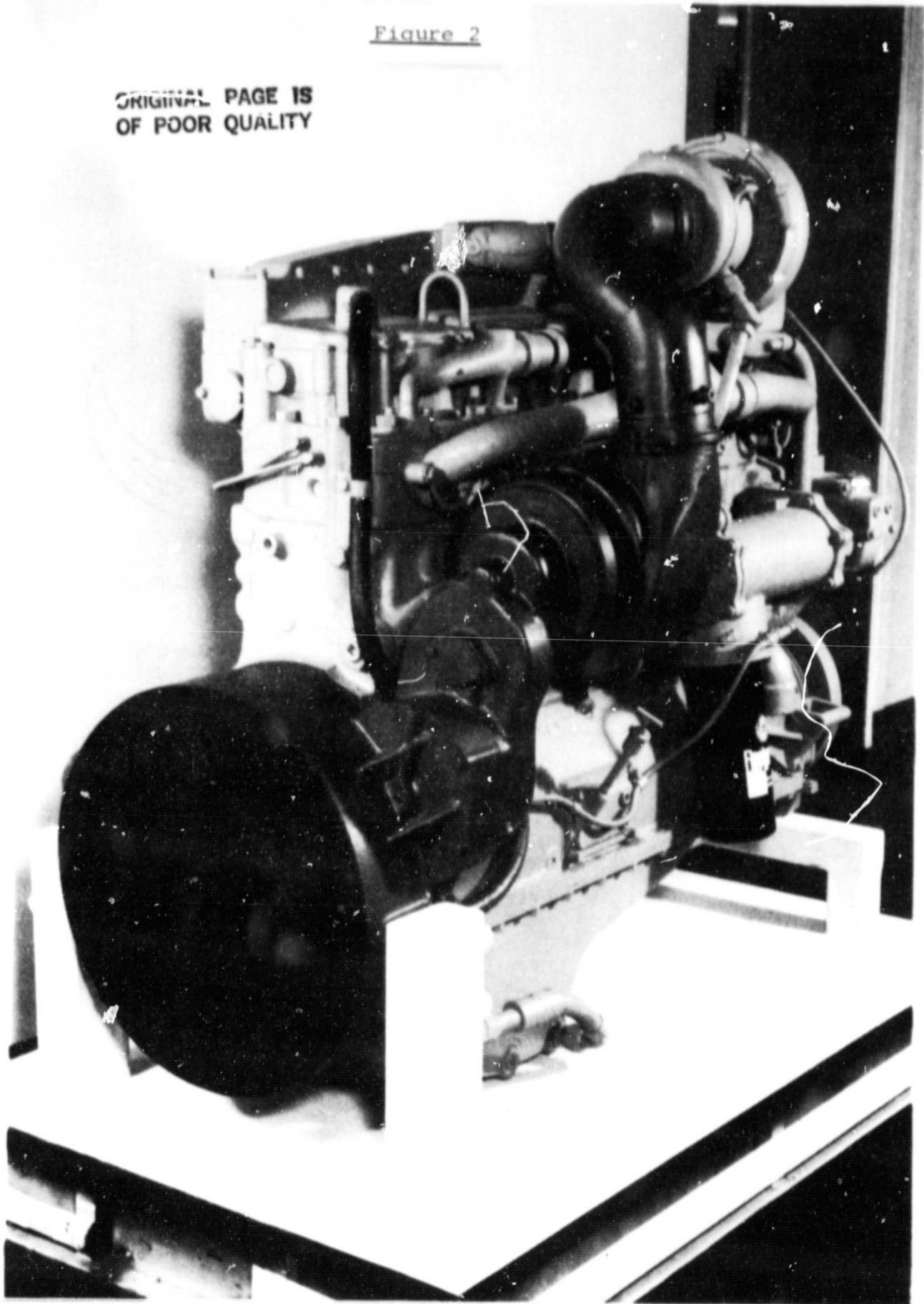


Figure 2

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CUMMINS NH TURBOCOMPOUND DIESEL ENGINE

The bushing seizure was caused by the fluid coupling shaft bottoming in the gear housing bushing bore due to improper stackup of the shaft-fluid coupling assembly. Due to this problem, it was decided to update the test engine with a revised turbocompound gear train housing which corrected this problem and also eliminated external oil drillings for lubrication of the gears.

Steady-state performance mapping consisting of a matrix of engine speed and load conditions was then completed, along with measurement of gaseous emissions. Baseline data repeatability was achieved with a measured brake specific fuel consumption (BSFC) of .323 lb/bhp-hr at the California 6 gram combined ( $\text{BSNO}_x + \text{BSHC}$ ) gaseous emission level.

### 3.0 Engine Performance Testing and Upgrading: Task II

A new cast exhaust manifold was designed to improve the pulse conservation of the exhaust blowdown. The junctions at each port connection were designed to maintain a constant area such that the exhaust pumping work was minimized. This manifold provided BSFC improvements of .003 lb/bhp-hr at rated power and a .005 lb/bhp-hr at torque peak power. The most significant BSFC improvements were made at part load in the 1300-1600 rpm engine speed range.

In an effort to optimize the exhaust valve timing, a new exhaust camshaft lobe was designed which opened the exhaust valve  $10^\circ$  earlier and had  $10^\circ$  longer dwell than the baseline cam. Engine performance testing did not show any fuel consumption improvement with the new cam but did show an increase in available exhaust gas energy. The engine performance may improve however, as the turbomachinery efficiency increases such that the additional blowdown energy converted to turbomachinery work is greater than the reduced in-cylinder work.

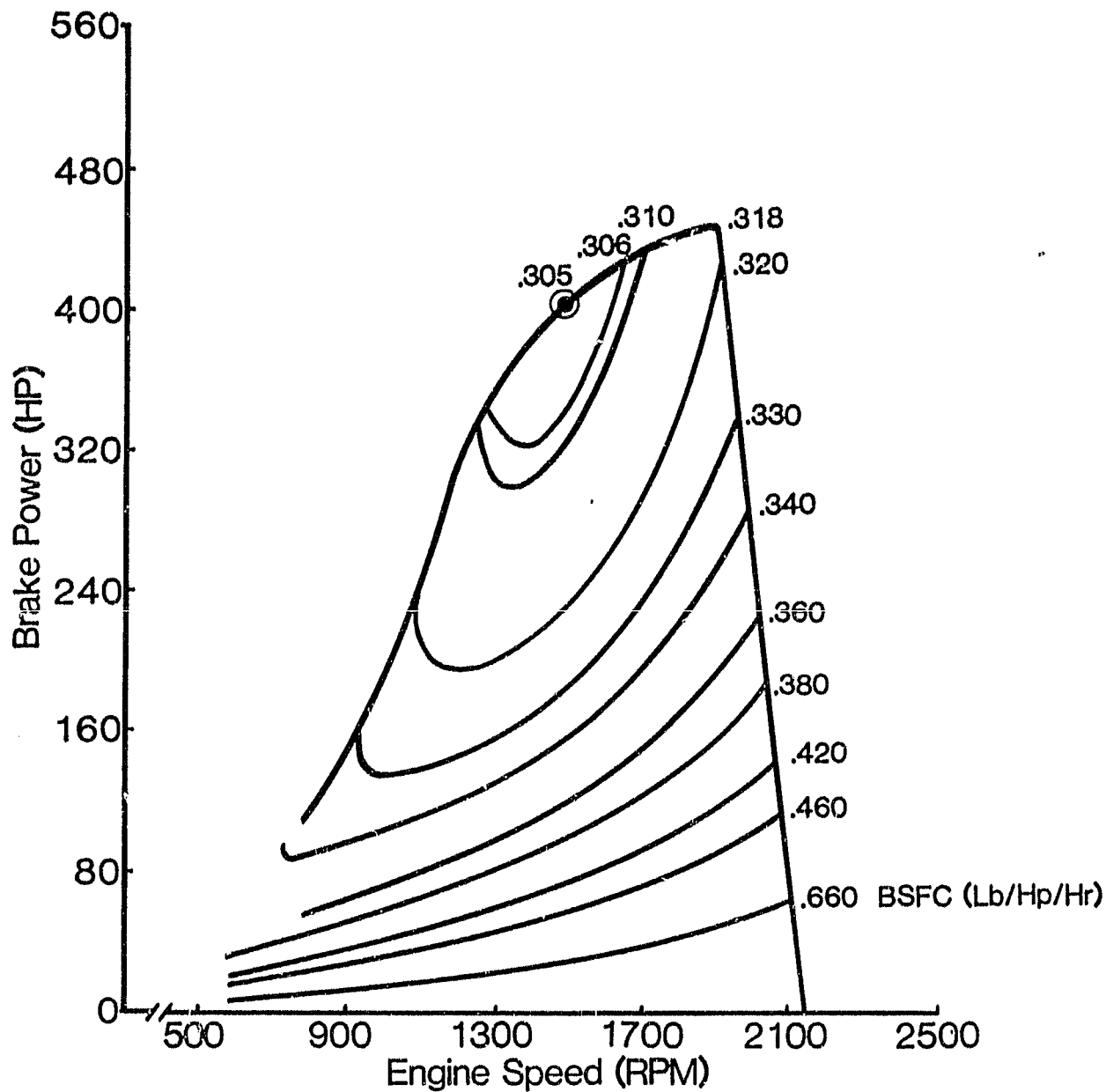
A new injector camshaft lobe was designed to provide improved injection characteristics for the advanced turbocompound engine. The new cam reduced the injection duration by  $0.5^\circ$  crank angle and increased the injection pressure by approximately 3000 psi over the interim baseline cam. The higher injection pressure and resultant shorter duration accelerates the air-fuel mixing rate. Performance testing at the same emission level resulted in a .002-.003 lb/bhp-hr BSFC reduction along the torque curve.

At this point, a performance map was generated which showed a BSFC achievement of .318 lb/bhp-hr at rated power, .307 lb/bhp-hr at torque peak power, and a minimum of .305 lb/bhp-hr at 1500 rpm. The fuel map is shown in Figure 3. It should be noted that while a gain of .005 lb/bhp-hr was achieved over the interim turbocompound engine at rated power, significantly larger BSFC gains were achieved at part load and at the lower engine speeds (1300-1600) where an engine would typically operate on a level road at 55 mph.

A 13-mode gaseous emission cycle was also conducted to determine the injection timing required to conform to the California 6

Figure 3

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UPGRADED TURBOCOMPOUND ENGINE  
OPERATING SPEED AND LOAD RANGE  
NOVEMBER, 1980

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gram (BSNO +BSHC) gaseous emission limit. The mechanical variable timing was<sup>x</sup> set at 14° BTDC dynamic timing for normal operation and advanced to 21° BTDC dynamic timing during light load operation. This produced a 5.86 gm/bhp-hr combined (BSNO<sub>x</sub>+BSHC) gaseous emission level.

The new performance map was put into Cummins' Vehicle Mission Simulation (VMS) computer program to ascertain the tank mileage improvement over the interim turbocompound engine. The VMS program input requirement consists of a detailed description of the vehicle and selection of a route. The model is capable of adjusting to varying ambient operating conditions such as temperature plus prevailing wind velocity and direction. The VMS can predict both steady-state performance and transient engine behavior.

Output data under steady-state operating conditions includes startability, gradeability, and vehicle performance in all the transmission gears. The route simulation summary included trip time, average speed, fuel consumption, gear shifts, time spent at full throttle, and average engine load factor.

A high degree of confidence in the predictive accuracy of the VMS model was achieved by comparing the interim turbocompound vehicle test results with the calculated results of VMS. Therefore, the VMS program was utilized for predicting performance gains of the advanced turbocompound engine.

A VMS run was made for the Pilot Center fuel economy route at the completion of Task II. This route consists of public roads beginning at the Cummins Technical Center in Columbus, Indiana, going south through Louisville, Kentucky, turning east to Cincinnati, Ohio, and returning to Columbus. VMS predicted a tank mileage of 5.59 mpg for the upgraded engine versus 5.40 mpg for the interim engine, or a 3.5% upgraded turbocompound engine tank mileage improvement.

#### 4.0 Advanced Engine Preparation: Task III

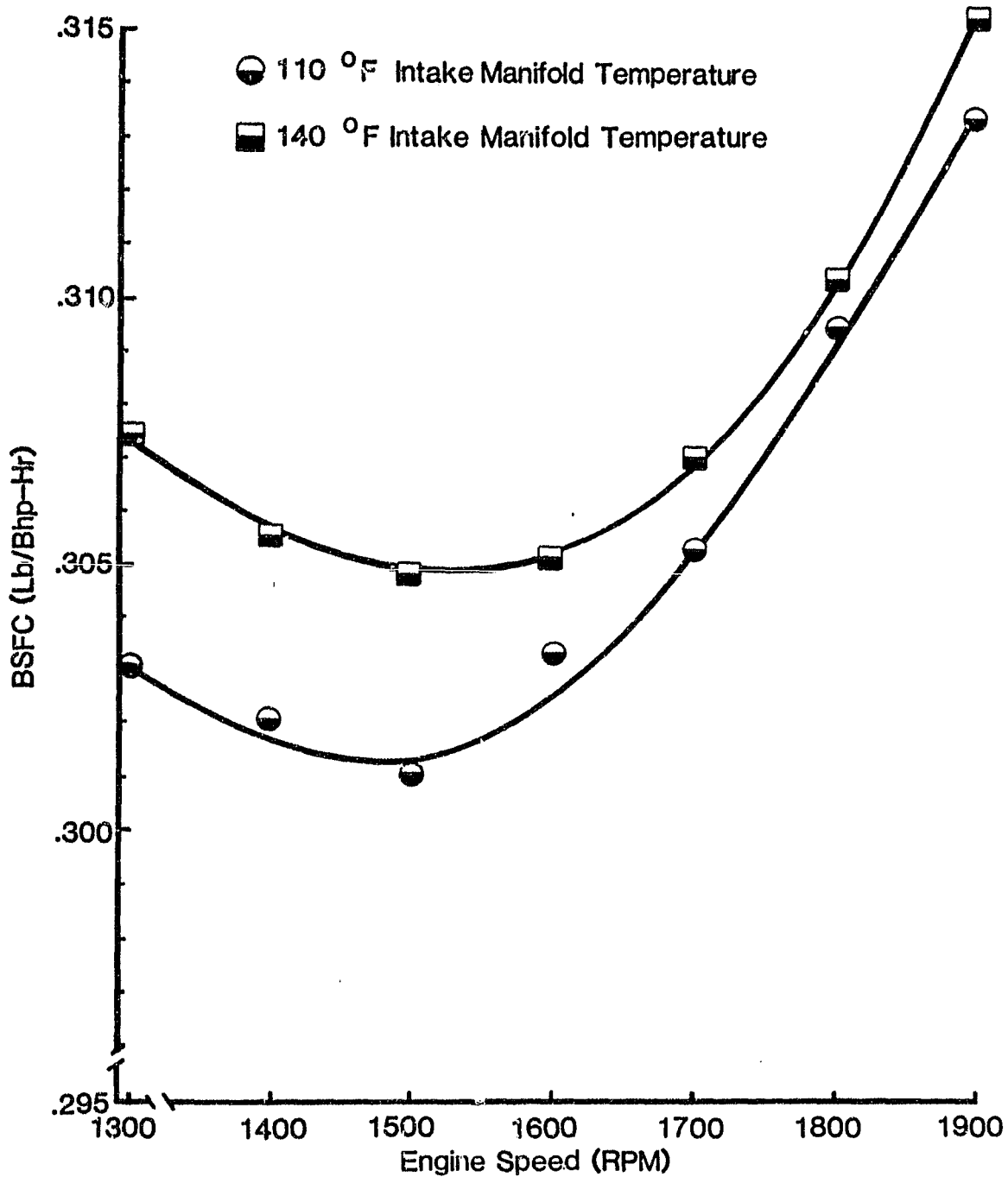
##### 4.1 Engine System Optimization

The engine cooling system during the vehicle tests utilized a two-pump, two circuit cooling system providing a 140°F intake manifold temperature on an 85° day at rated power. A further reduction in intake manifold temperature to 110°F is possible with a chassis mounted air-to-air aftercooling system. A Class VIII truck engine currently in production utilizes this type of cooling system.

The comparison between engine fuel consumption along the torque curve for intake manifold temperatures of 140°F and 110°F is seen in Figure 4. The fuel consumption benefit of .001-.004 lb/bhp-hr was due to an increase in air-fuel ratios, resulting in improved combustion efficiency. The air-fuel ratio changed from 24.8 to 26.3 at torque peak power with 110°F IMT while the air-fuel ratio changed from 30.1 to 30.5 at rated power.

Figure 4

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INTAKE MANIFOLD TEMPERATURE COMPARISON  
TORQUE CURVE - BSFC VS. ENGINE SPEED

Exhaust manifold optimization, in addition to the Task II effort, was attempted utilizing an in-house program which simulates the exhaust process. With this program, the exhaust gas undergoes a blowdown loss as the exhaust valve opens and the gases flow from the cylinder through the valve opening and into the exhaust manifold. In addition, pumping work occurs as the piston moves upward to force the exhaust gases into the exhaust manifold. The manifold is evaluated as a straight duct with a modified friction coefficient to account for the bends in the manifold. The optimum inside diameter (I.D.) is established by comparing the trade-off between the increased blowdown losses at the valve as the manifold I.D. increases, the increased pumping work as the manifold I.D. decreases, and the amount of work extracted from the exhaust energy through the turbocompound turbine and gear train. This trade-off curve is seen in Figure 5. The optimum inside diameter for the exhaust manifold appears to be in the region of 1.6 to 1.8 inches.

In an effort to optimize the exhaust manifold I.D., a manifold of 1.6 inches in cross sectional diameter was evaluated (the existing manifold was 1.8 inches in diameter). Performance testing was completed for both of these cast exhaust pulse manifolds. The results are shown in Figure 6. The 1.6 inch diameter manifold had a .001 to .002 lb/bhp-hr increase in brake specific fuel consumption along the torque curve. This indicated that the optimum exhaust manifold I.D. was nearer 1.8 inches. To fully substantiate this conclusion, a 2.0 inch I.D. manifold should be evaluated; however, it was not pursued during this program.

Insulation of the exhaust system to provide increased available exhaust gas energy to the turbines was evaluated. The externally applied insulation consisted of an alumina-silica refractory fiber blanket. The alumina-silica material chosen was suitable for continuous exposure to 2400°F in a normal oxidizing atmosphere.

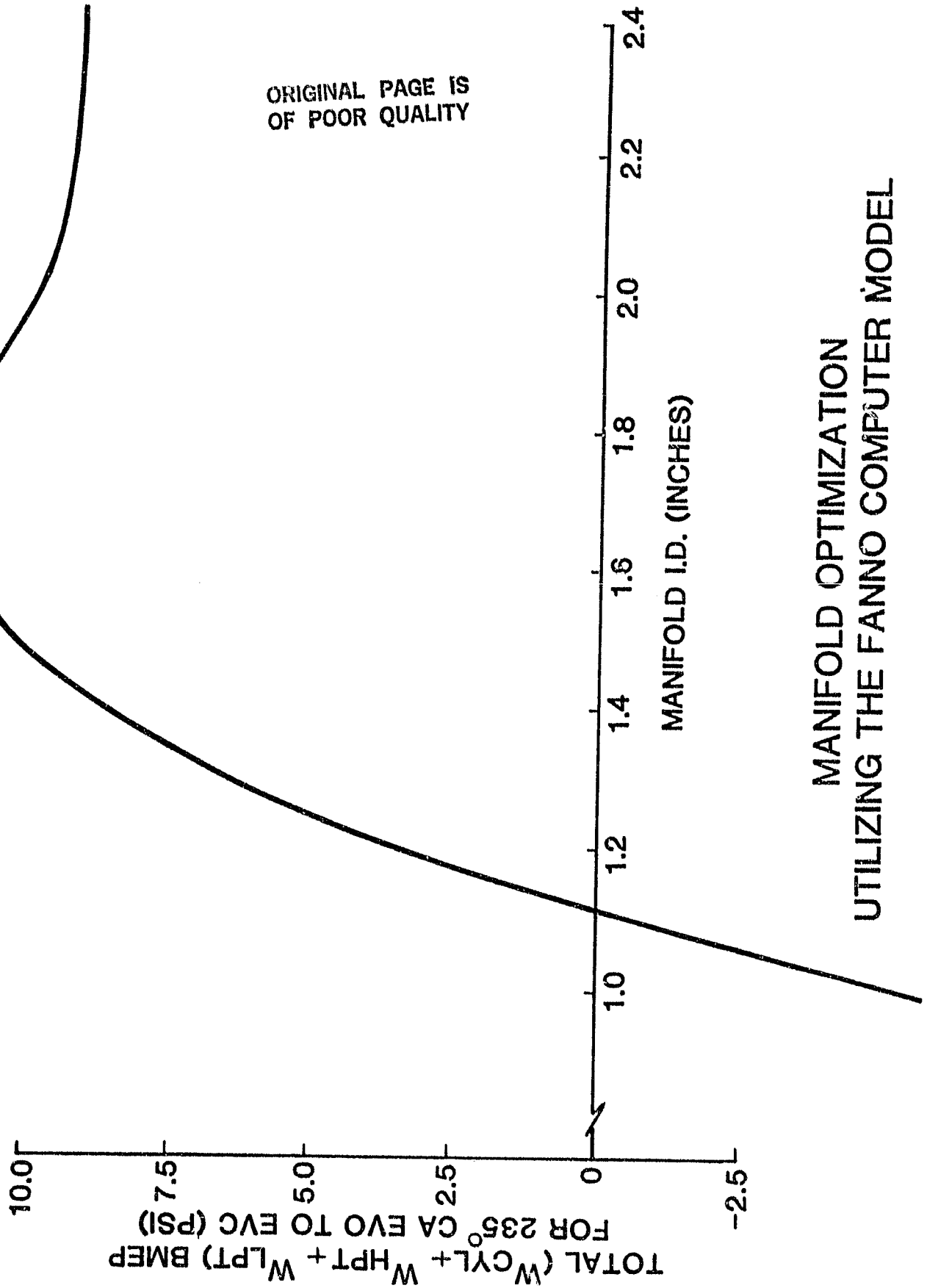
Insulated versus non-insulated engine performance testing was completed using the blanket to externally insulate the exhaust manifold, charge air turbine volute, and interstage duct. The temperature increase at the charge air turbine was 10°F while the power turbine inlet showed an increase of 15°F. This was due to the cumulative effect of heat loss reduction over the exhaust manifold, charge air turbine volute, and interstage duct. The increase in exhaust gas enthalpy provided a reduction in fuel consumption of approximately .002 lb/bhp-hr along the engine's torque curve. The fuel consumption for the turbocompound engine is shown in Figure 7 for the insulated and non-insulated configurations.

#### 4.2 Free Power Turbine

The Phase III turbocompound power turbine shaft was supported by semi-floating journal bearings. The journal bearing power turbine was tested on the Cummins turbine map stand to evaluate shaft bearing losses. Bearing losses were measured by determining the heat rejection to the oil from the bearing housing. This is accomplished by accurately measuring the oil flow to the bearing housing. Heat transfer effects were minimized by using a turbine



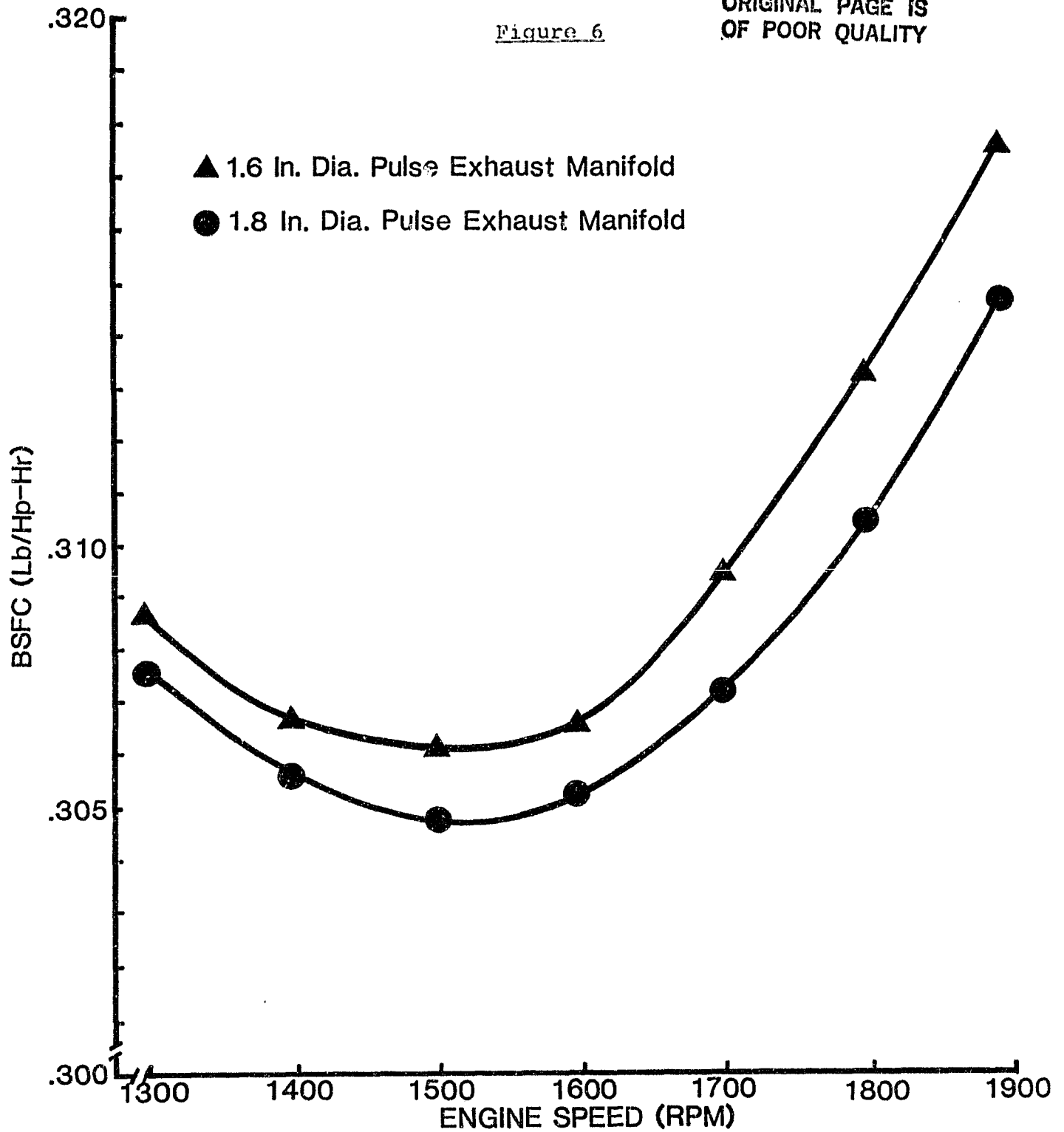
Figure 5



MANIFOLD OPTIMIZATION  
UTILIZING THE FANNO COMPUTER MODEL

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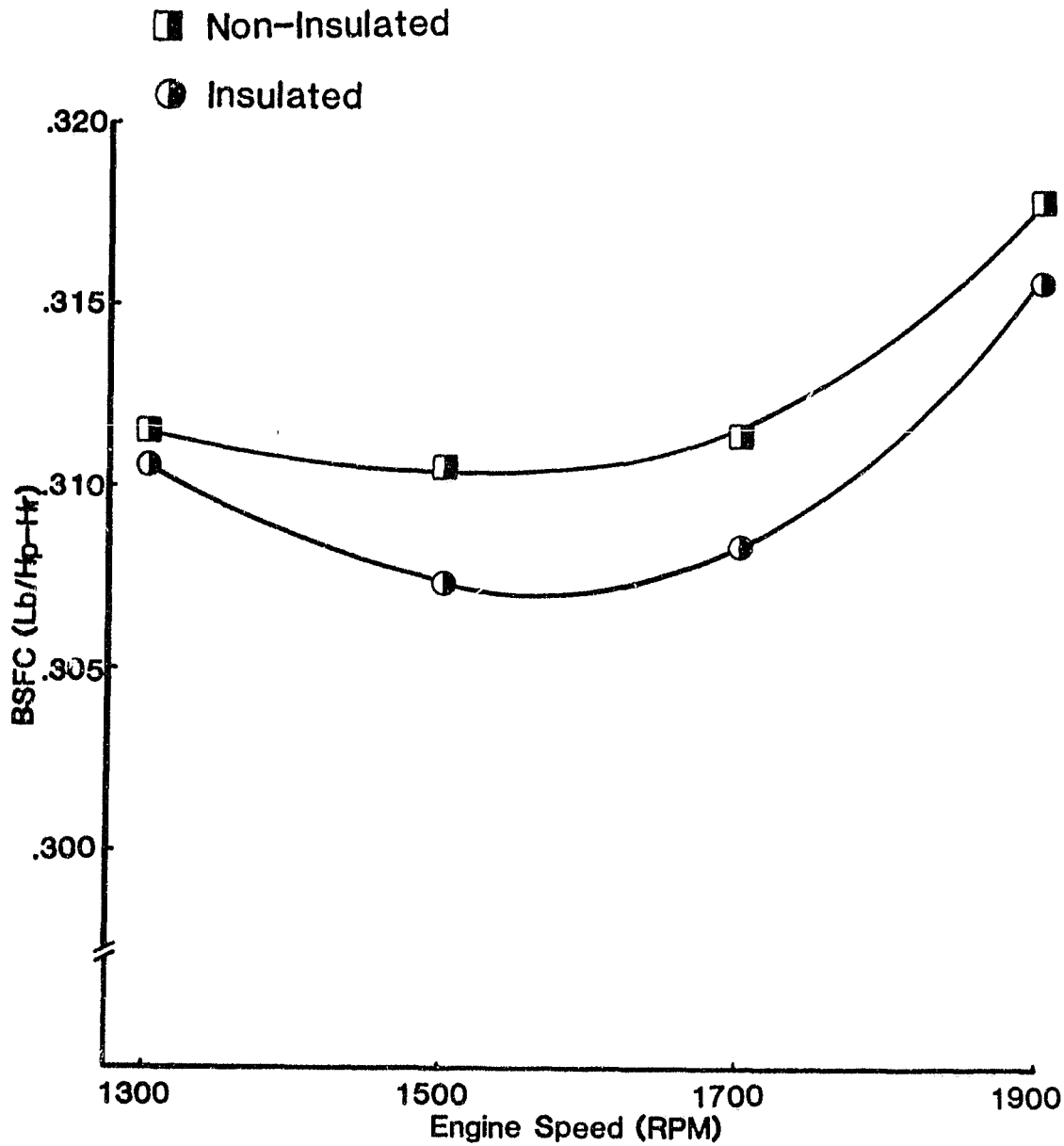
Figure 6



EXHAUST MANIFOLD COMPARISON  
TORQUE CURVE - BSFC Vs. ENGINE SPEED

Figure 7

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TCPD-450 PERFORMANCE DATA  
INSULATED Vs. NON-INSULATED EXHAUST SYSTEM

inlet temperature which was approximately 250°F resulting in a minimum temperature differential between the test unit and the incoming oil.

A prototype ball bearing supported power turbine shaft was designed to determine the potential reduction in shaft mechanical losses compared to the journal bearings. The ball bearing arrangement incorporated spring loaded, angular contact ball bearings. The design used a modified aluminum bearing housing with an increased bore to accommodate the larger bearing carrier. The bearing carrier is steel and has radial clearance within the housing to provide oil film damping. The carrier also incorporates lube orifices which target a jet of oil on the inner race of each bearing. The lube jet method requires accurate targeting of the flow to get oil into the bearing working against the aerodynamic resistance of the spinning balls and cage. The oil is then carried through the balls and into the bearing housing cavity. The center cavity is drained or vented to the housing by holes in the bottom of the carrier. A pair of stacked wavy washers is used to provide axial preload. The inner races were fixed to the shaft against rotation by an axial clamp from the high speed pinion nut. The unassembled unit is shown in Figure 8. The benefit to be gained through the utilization of ball bearings includes not only a reduction in friction, but also a possible improvement in turbine efficiency made possible by a reduction in operating clearances due to a stable shaft orbit.

After the initial journal bearing arrangement was evaluated, the turbine rotor and shaft were modified by reducing the diameter to accept the ball bearings. The comparison test of the ball bearing power turbine was performed using the same power turbine, turbine rotor load compressor, and rotor to shroud clearances.

A summary of the friction loss data (as measured by heat rejection to the oil) is depicted in Figure 9. A graph of the ball bearing versus journal bearing mechanical efficiency is shown in Figure 10. Figure 9 also shows the parasitic loss in horsepower of the journal bearing and the ball bearing unit at rated and peak torque speeds. This difference in horsepower is available as shaft power for the engine.

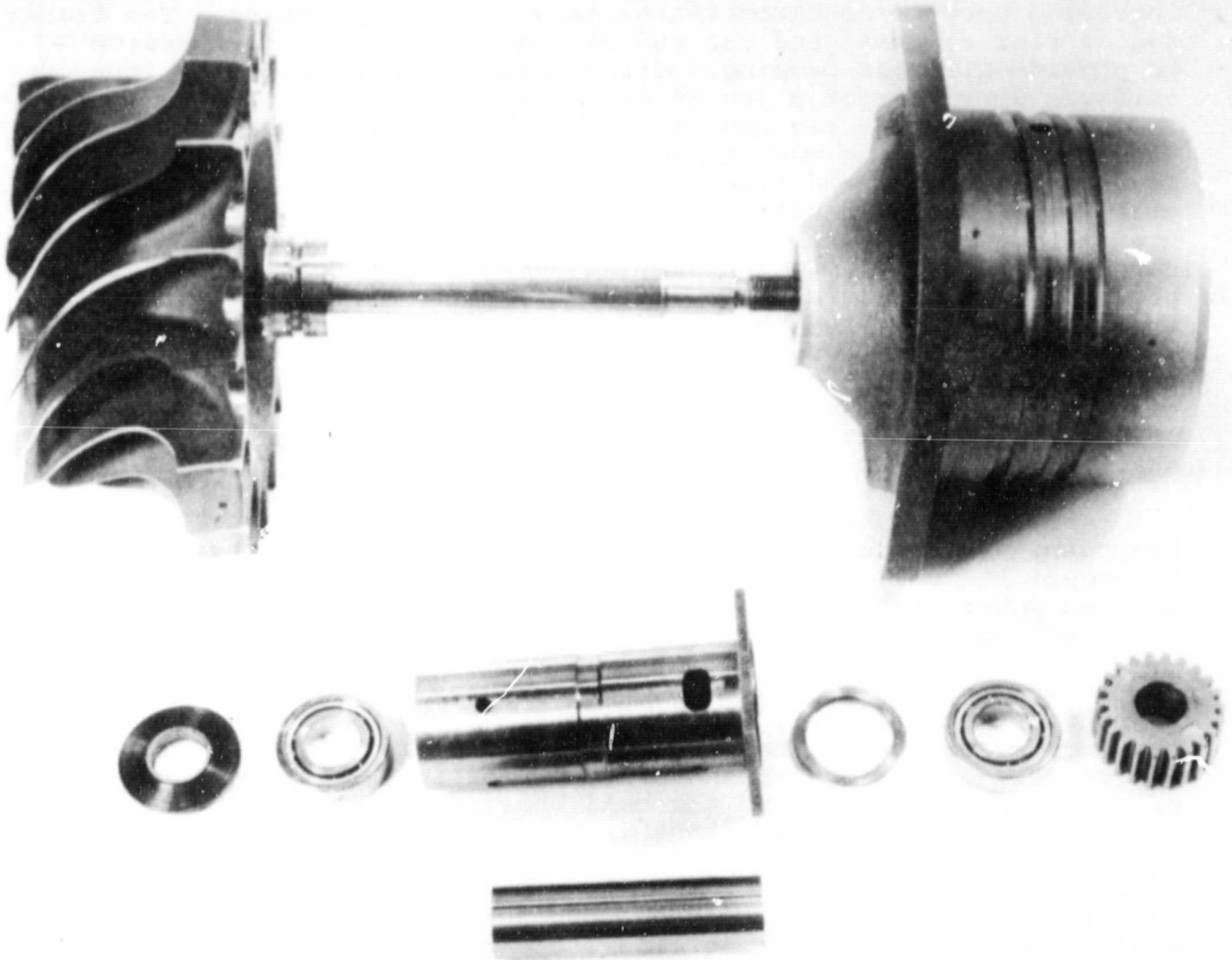
Further testing was to have been carried out to evaluate the gains from reduced operating clearances made possible by the ball bearing system, however, bench testing indicated that the shaft stiffness was not sufficient to avoid a flexural mode. This type of motion would not allow an evaluation of the ball bearing power turbine with reduced clearances. A second design iteration would be required to provide sufficient bearing life, satisfactory shaft dynamics and oil film damping, minimum shaft flexing, and consideration for cost and complexity. A second design iteration was not pursued due to the design and procurement lead times.

#### 4.3 Power Turbine Speed Optimization

In an effort to improve the aerodynamic efficiency of the power turbine, an increase in operating speed was required. This

Figure 8.

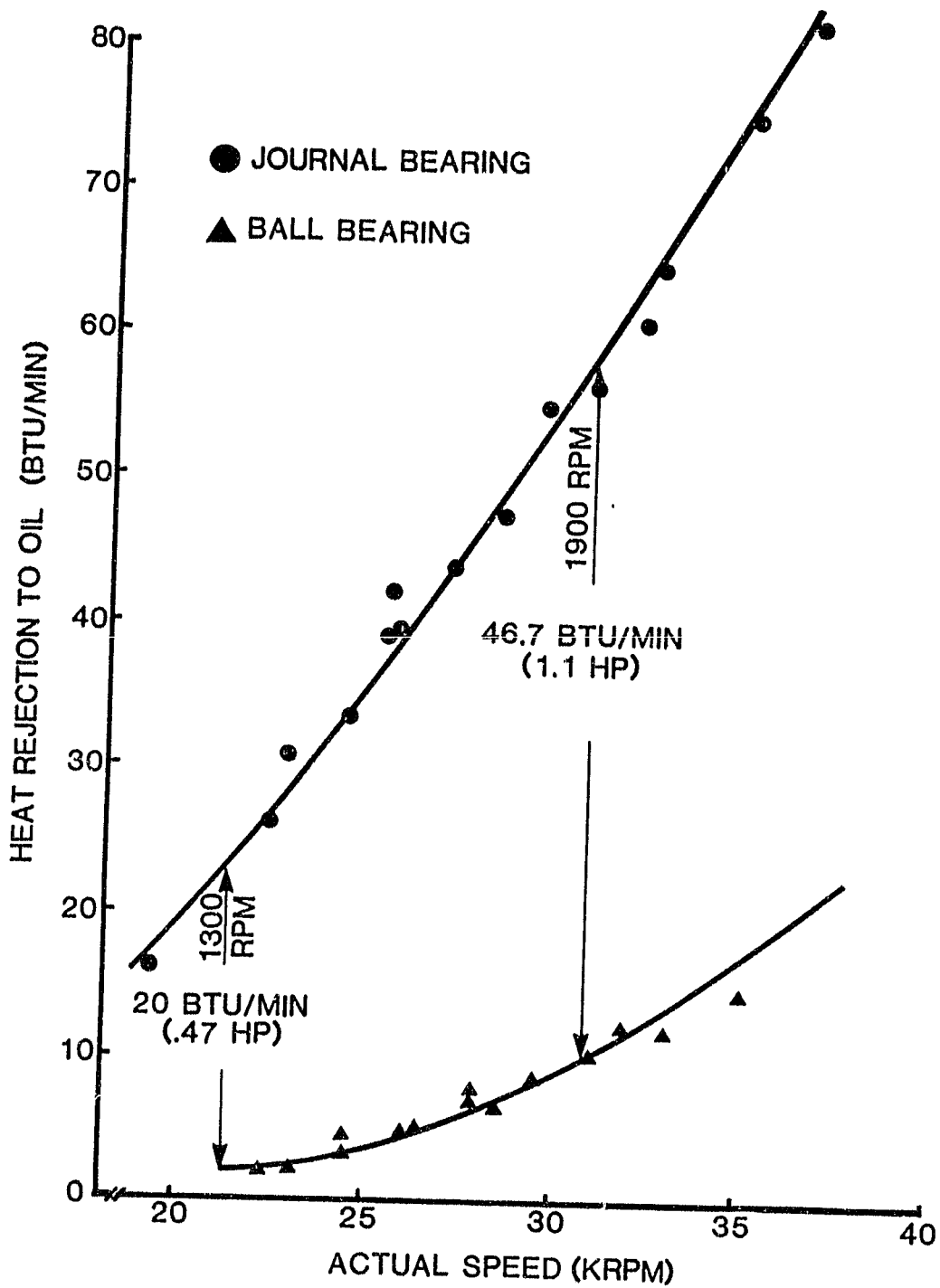
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BALL BEARING POWER TURBINE ARRANGEMENT  
EXPLODED VIEW

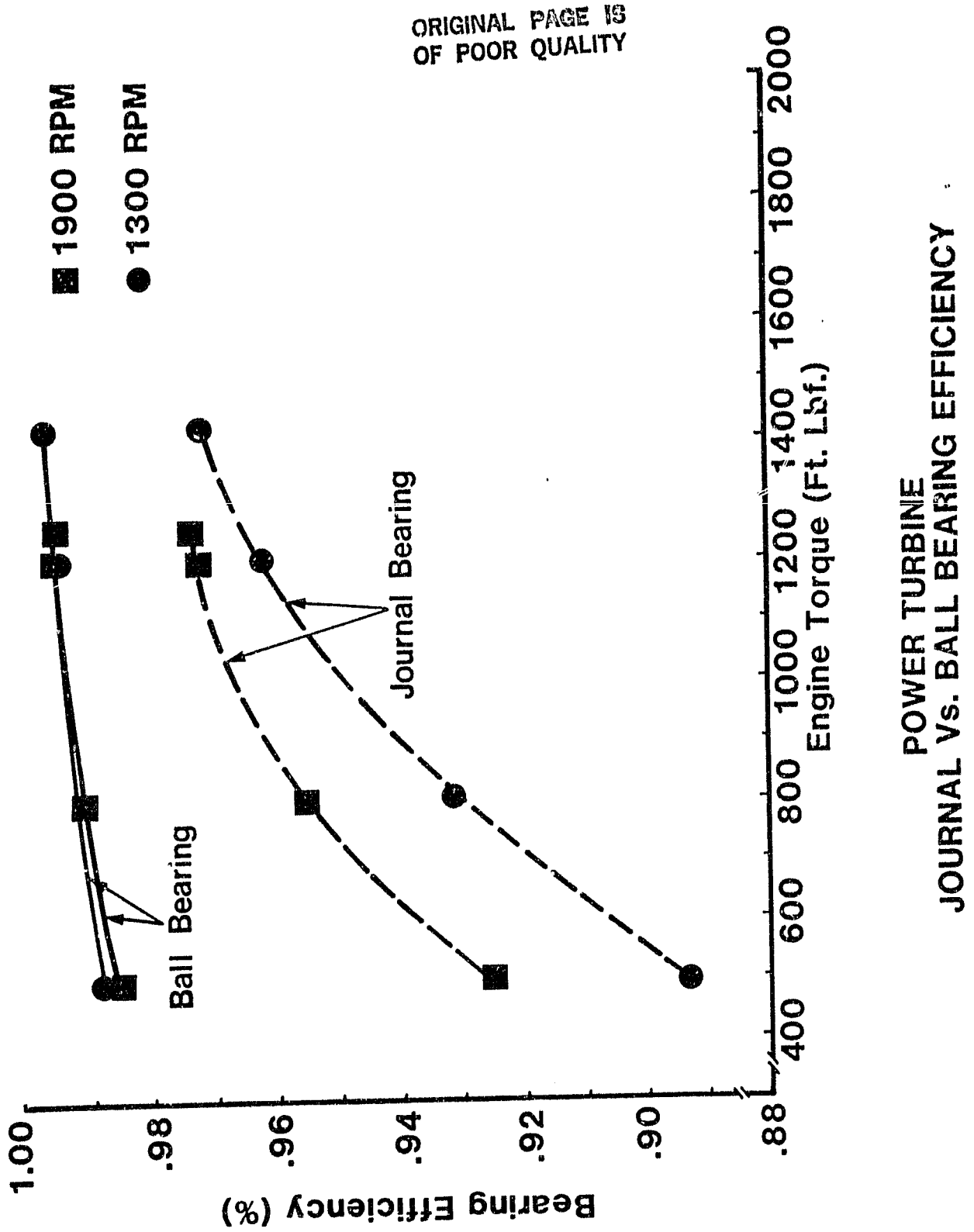
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Figure 9



POWER TURBINE BEARING  
HEAT REJECTION TO OIL

Figure 10



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was accomplished by varying the gear ratio with resultant changes in the operating line of the turbine shown in Figure 11. The interim turbocompound power turbine operated at a gear ratio of 15.78 times the engine speed. Operating lines are shown for engine speeds of 1900 and 1300 rpm at three gear ratios: 15.78, 16.42, and 17.12 times the engine speed. At 1900 rpm, as the gear ratio is increased the power turbine efficiency remains relatively constant at full load, but decreases slightly at part loads. At 1300 rpm, however, the power turbine efficiency improves significantly over the full operating range. Hardware was procured to increase the gear ratio to 16.42 and 17.12 times the engine speed.

Engine performance tests were completed with all three gear ratios throughout a matrix of engine speed and load conditions. The intermediate ratio of 16.42 was selected as the optimum ratio after evaluation of the full and part load fuel consumption data shown in Figure 12 and Figure 13. An average fuel consumption benefit of .001 lb/bhp-hr was measured along the torque curve over the interim gear ratio of 15.78. This is consistent with the predicted benefits at full load using the improved efficiencies shown in Figure 11.

#### 4.4 Abradable Shroud Turbine

As stated previously, Task III involved applying practices and techniques typically employed in gas turbine power plants into the turbocompound engine turbomachinery. One of these practices for increased efficiency levels is the utilization of abradable materials in order to reduce clearances between the rotor and shroud.

An abradable nichrome/polyester composite coating was applied to the charge air turbine shroud. The nichrome/polyester coating has a temperature capability of 1500°F. The abradable turbine shroud was initially evaluated on the engine. Performance testing was completed for the abradable versus baseline turbine shrouds. The cold clearance for the turbines evaluated are shown below:

	<u>Axial Clearance</u>	<u>Radial Clearance</u>
Baseline	.030	.010
Abradable	.015	(Minimum)

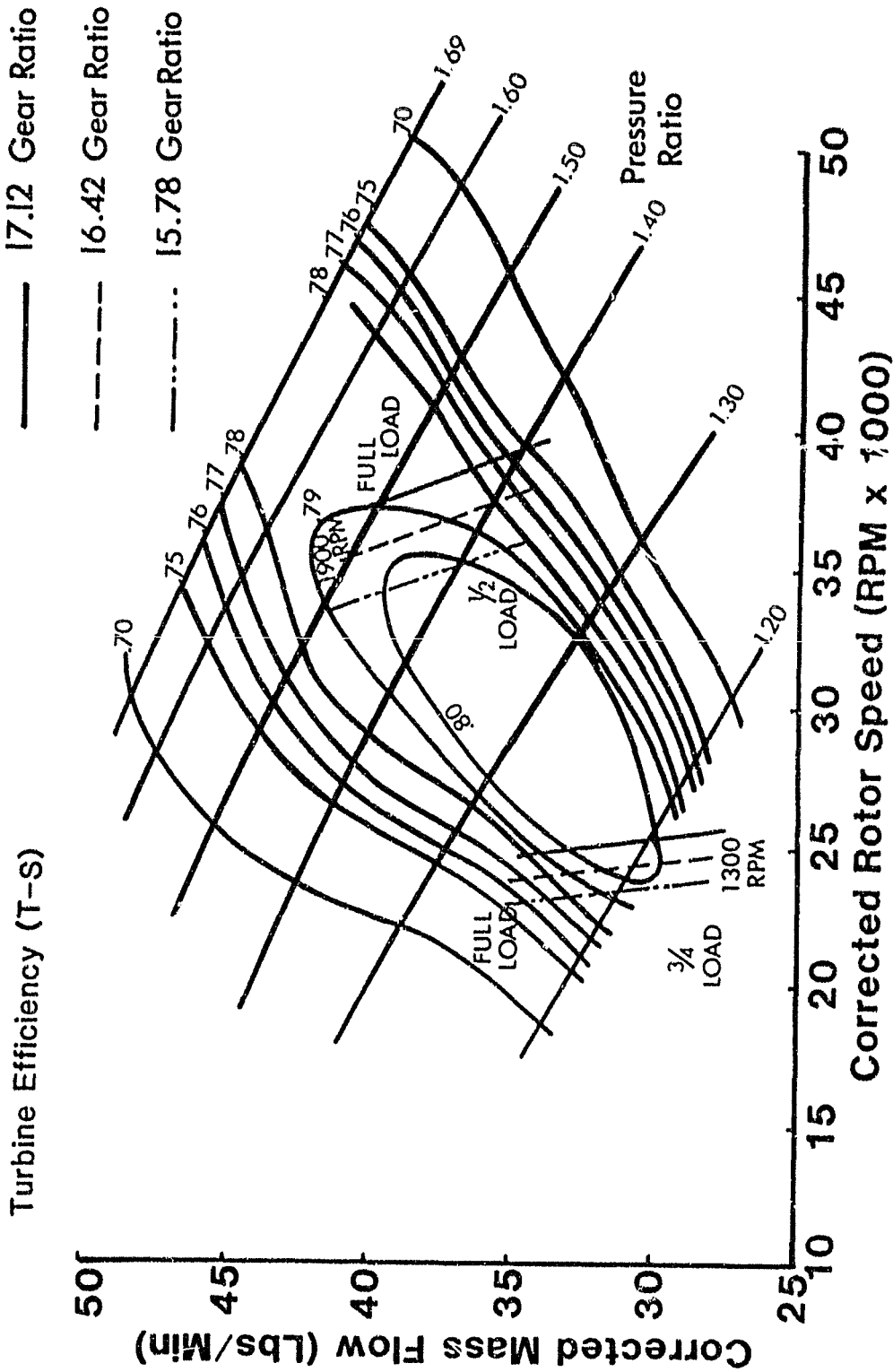
Test data along the torque curve did not indicate any fuel consumption advantage for the abradable turbine shroud.

Bench testing was subsequently completed to more accurately quantify the effect of clearances on turbine efficiency. Two turbine volutes were evaluated: a baseline shroud with no abradable coating and the turbine shroud with the abradable nichrome/polyester coating. Axial clearances for the abradable shroud were varied from .027 inch to .010 inch with minimal radial clearance. The baseline turbine shroud axial and radial clearances were set at .027 inch and .010 inch, summarized as follows:



Figure 11

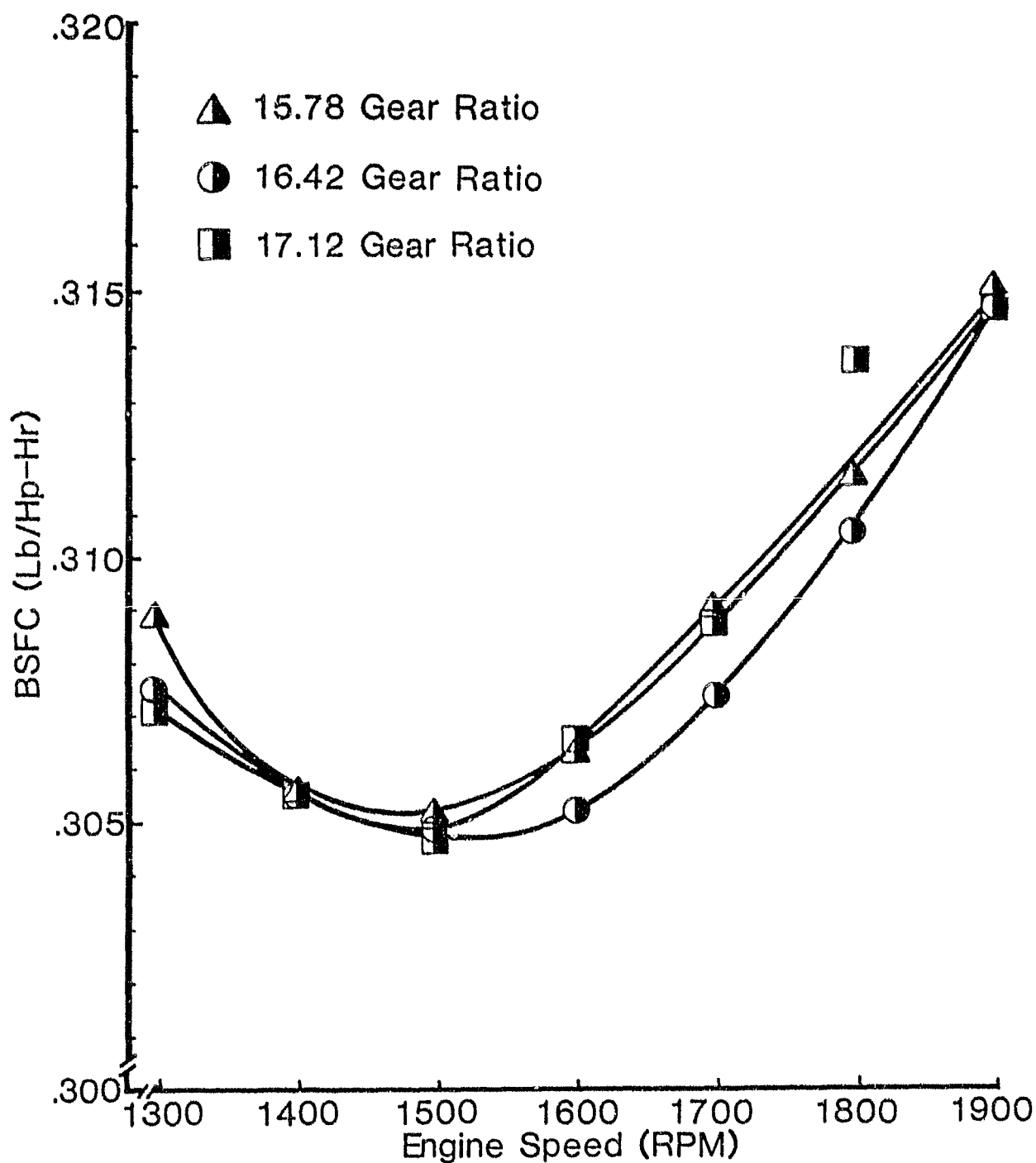
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# RADIAL POWER TURBINE EFFICIENCY MAP

Figure 12

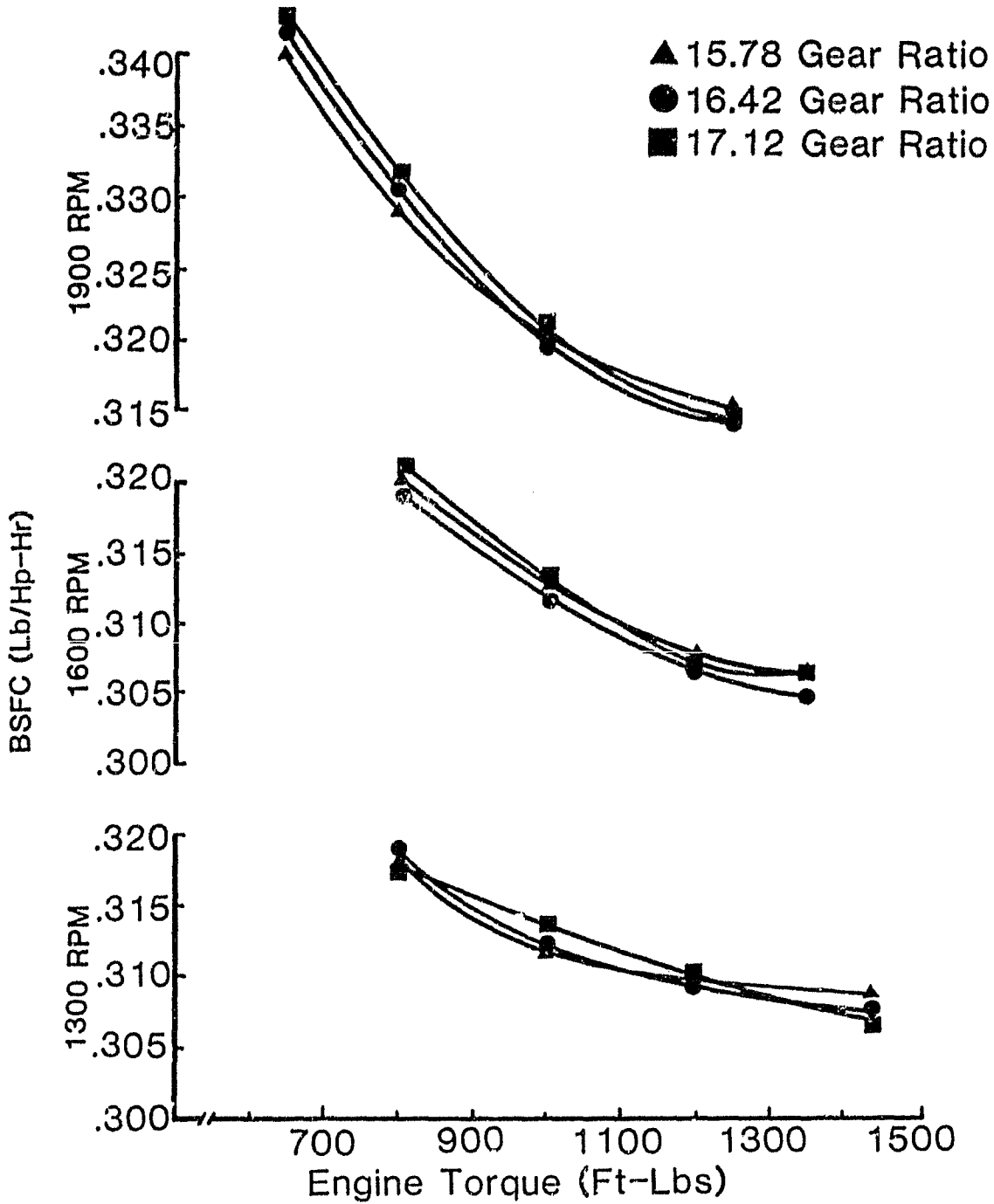
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GEAR RATIO OPTIMIZATION  
TORQUE CURVE - BSFC VS. ENGINE SPEED

Figure 13

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GEAR RATIO OPTIMIZATION  
BSFC Vs. ENGINE TORQUE

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	<u>Axial Clearance</u>	<u>Radial Clearance</u>	<u>Abradable Contour</u>	<u>Peak Turbine Efficiency</u>
Baseline	.027	.010	No	79.8
Build 2	.027	Minimum	Yes	80.9
Build 3	.018	Minimum	Yes	80.5
Build 4	.010	Minimum	Yes	80.3

The abradable charge air turbine shroud contour is shown in Figure 14 after the bench tests. The turbine efficiency versus rotor is shown in Figure 15. Taking into account test stand accuracy and repeatability, the abradable turbine shroud showed no difference in efficiency when the axial clearance was reduced from .027" to .010".

However, the data indicated a slight gain in turbine efficiency due to the lowering of the radial clearance. This improvement in performance, for this particular turbine, can be traced to the fact that a reduction in radial clearances reduces the blade to blade leakage in the area where a large portion of the work is being done.

#### 4.5 Abradable Compressor Shroud

For the compressor shroud, an aluminum-graphite composite coating was evaluated, as this material is compatible with an aluminum compressor rotor. The turbocharger compressor with the abradable shroud was first tested on the bench test stand with minimum radial clearance at the inducer region and .024 inch axial clearance between the rotor and the contour of the shroud near the exit. The axial clearance was reduced in .005 inch increments to a final .004 inch clearance at room temperature. The abradable shroud shown in Figure 16, after the last bench test, displays the coating abrasion, with some smearing, due to the compressor impeller. The performance data shown in Figure 17 shows the peak compressor efficiency increasing from 81.7 percent to 83.3 percent for the minimum clearance build and a 1-2% efficiency improvement at a constant pressure ratio. This illustrates both an improvement in peak efficiency and an increase in the width of the efficiency islands with use of the abradable compressor shroud. Thus, not only does the compressor work at higher efficiency levels, but it will also operate in these regions a greater percentage of the time.

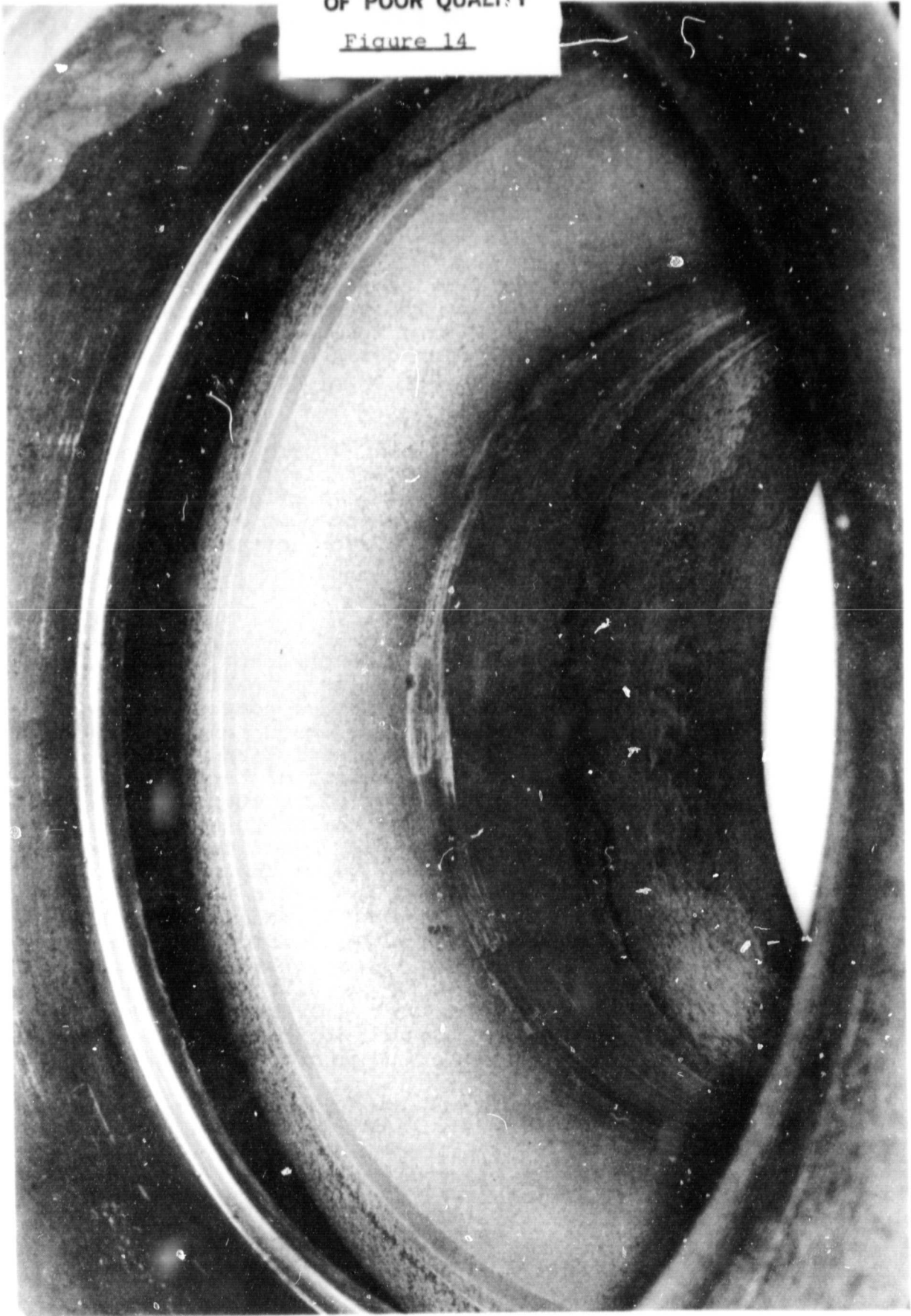
Engine performance testing was completed with the abradable compressor shroud at .024 inch and .004 inch axial clearance and minimum radial clearance. The engine test data shown in Figure 18 verified the 1-2% efficiency improvement measured on the bench test. A reduction in fuel consumption of .001-.003 lb/bhp-hr was measured along the torque curve.

#### 4.6 Abradable Heat Shields

In addition to abradable shrouds, heat shields for both the charge air turbine and power turbine were coated with feltmetal (R) abradable material to minimize the turbine backface clearance. The

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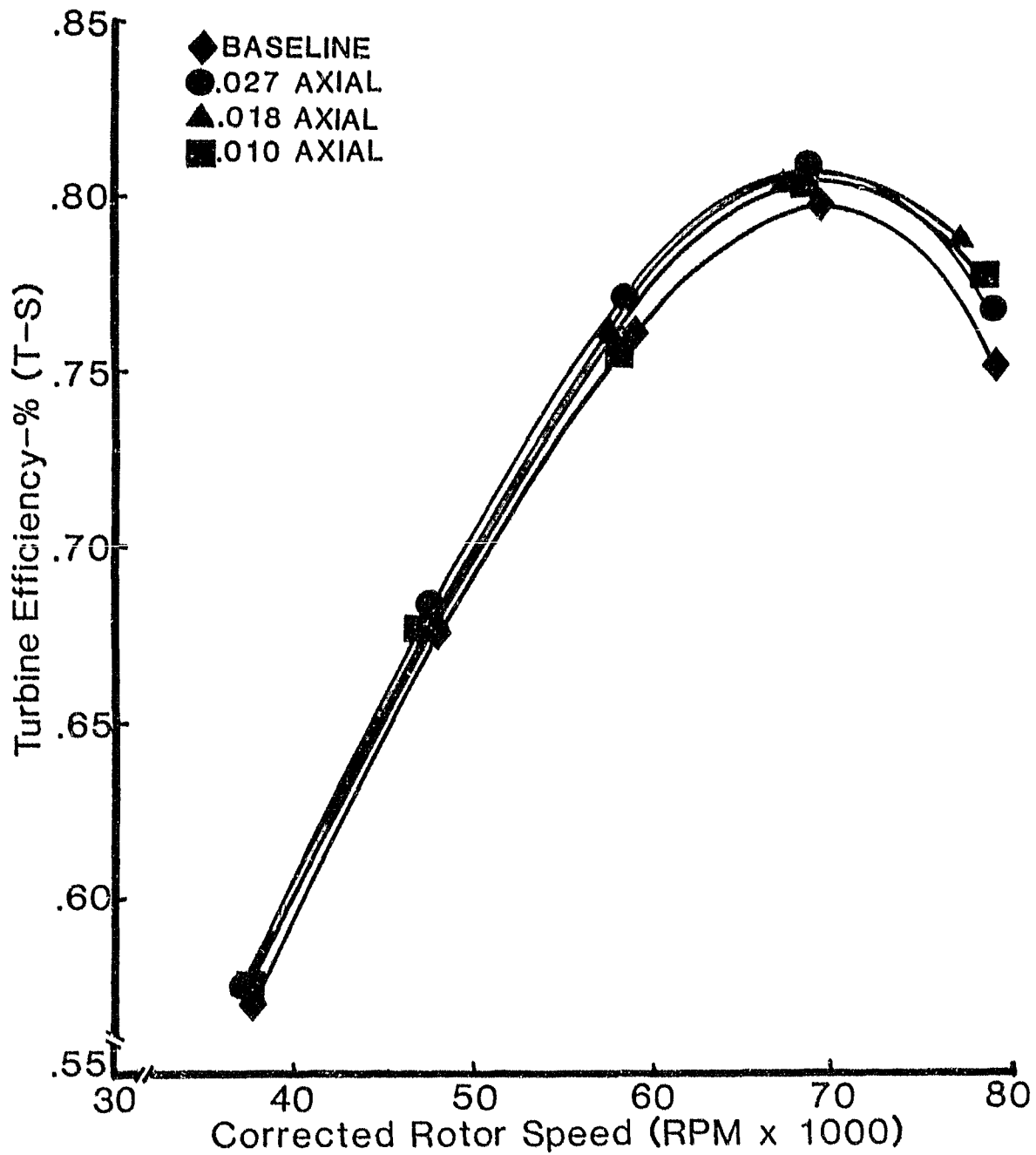
Figure 14



TURBINE ABRADABLE SHROUD

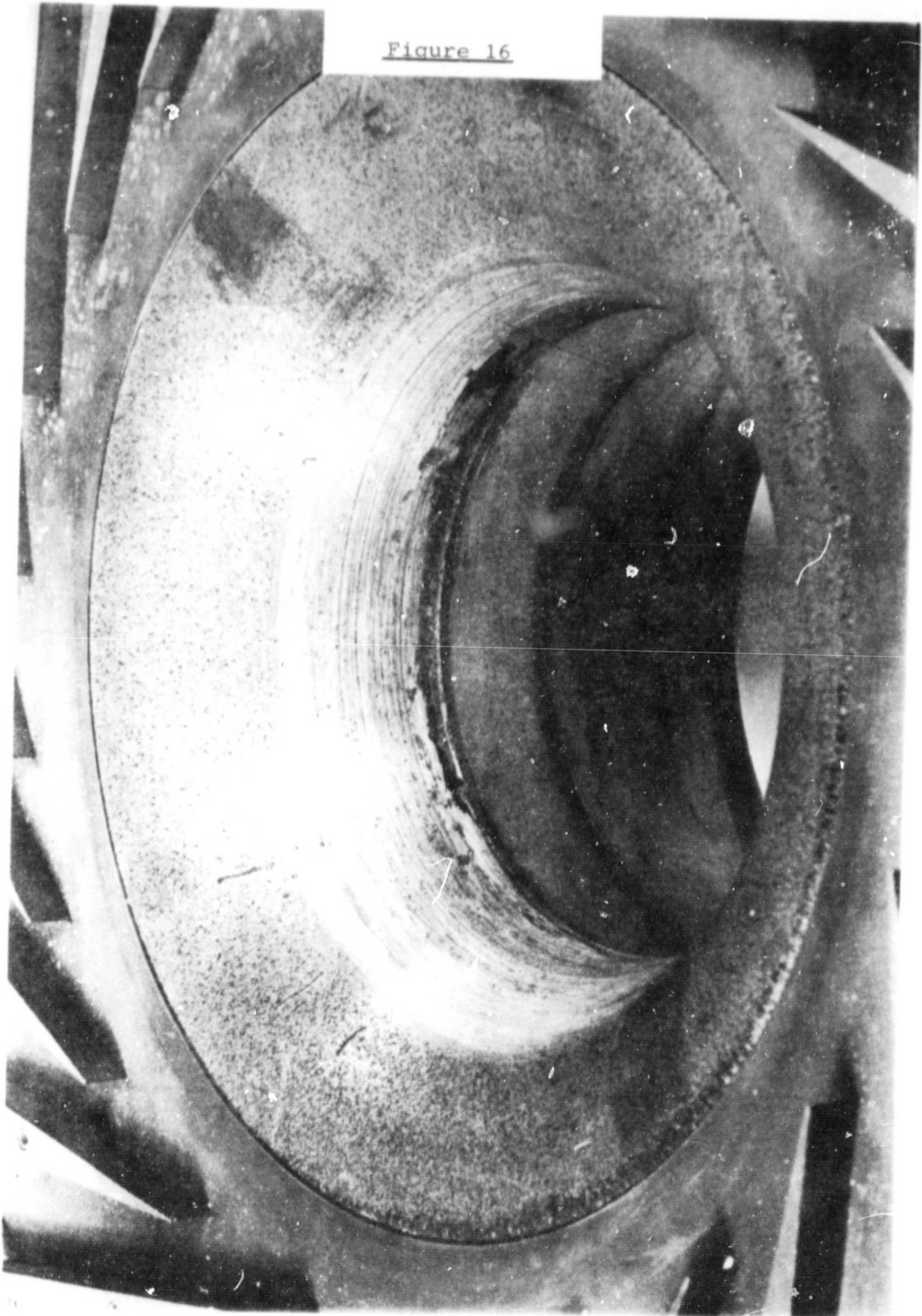
Figure 15

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TURBINE ABRADABLE SHROUD BENCH TEST  
PRESSURE RATIO 2.2

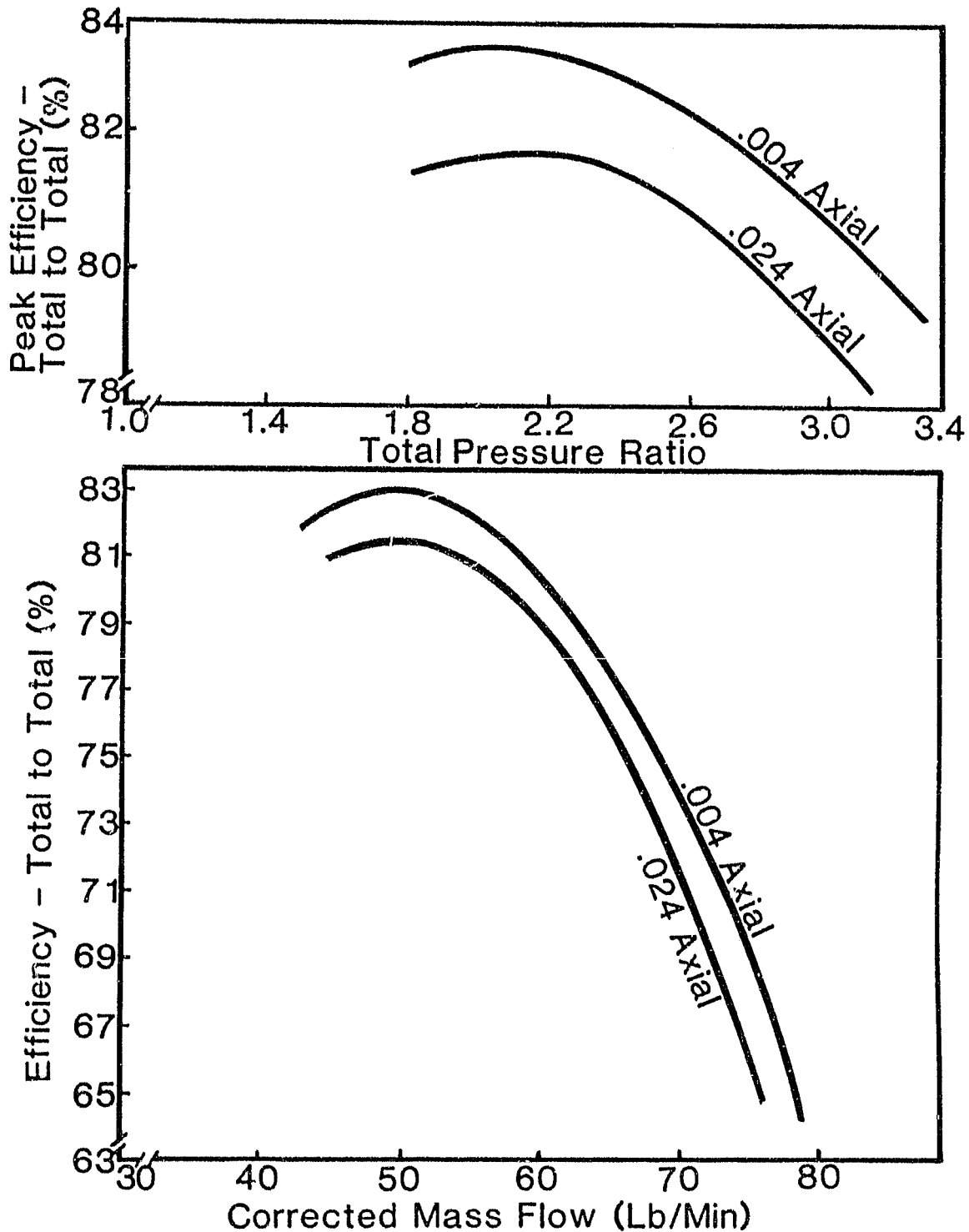
Figure 16



ABRADABLE COMPRESSOR SHROUD

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Figure 17

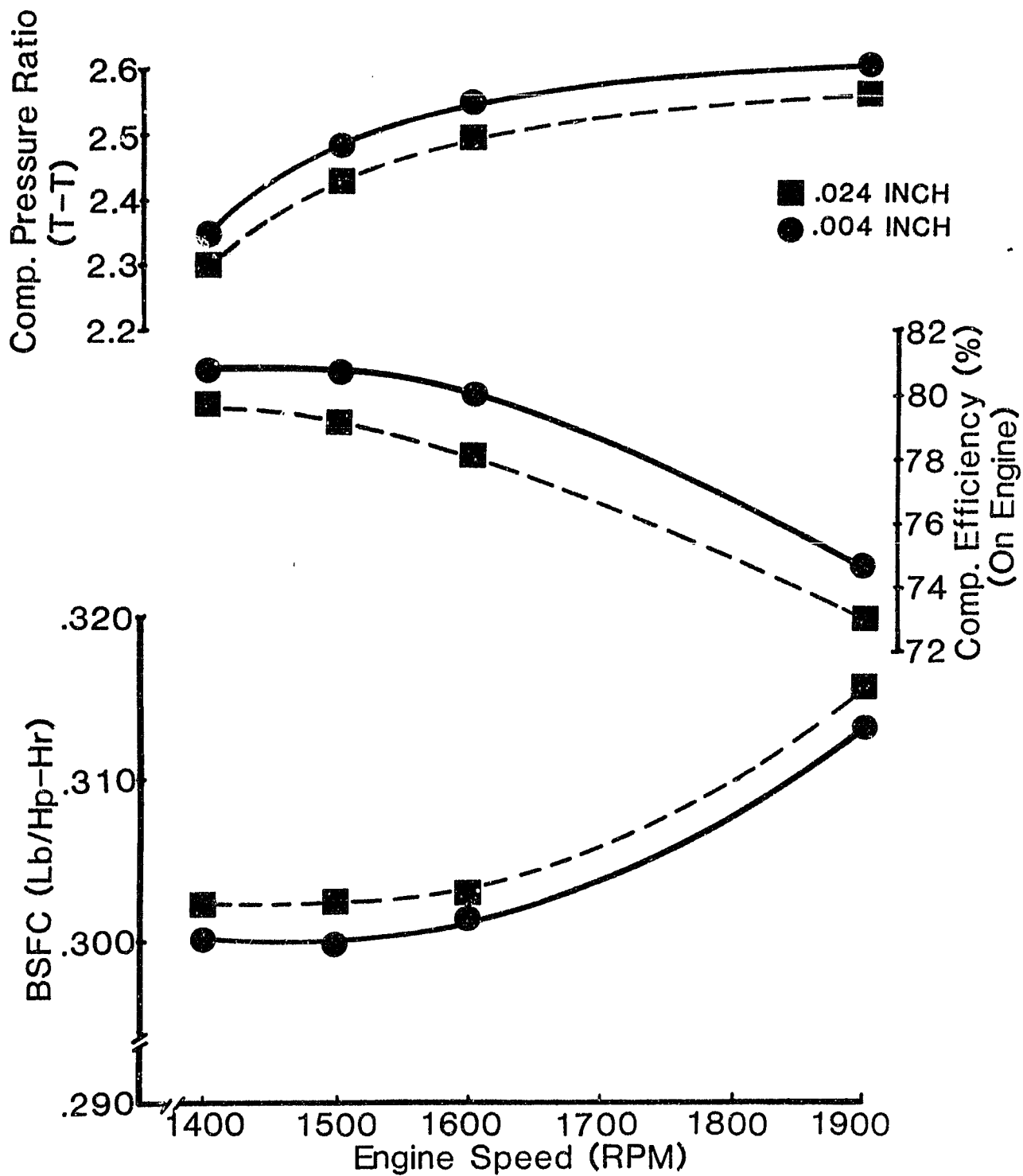


EFFECT OF CLEARANCE ON  
PEAK EFFICIENCY Vs. TOTAL PRESSURE RATIO AND  
COMPRESSOR EFFICIENCY Vs. MASS FLOW AT 2.2 P.R.



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Figure 18



TCPD-450 PERFORMANCE DATA  
ABRADABLE COMPRESSOR SHROUD

heat shield is located between the turbine rotor and bearing housing. Its primary functions are to reduce heat transfer to the bearing system and provide the rotor backface shroud.

Charge air turbine bench testing was completed with an abradable heat shield with minimum backface clearance and a baseline heat shield with .035 inch backface clearance. On the bench test stand, there was no turbine efficiency improvement with the abradable heat shield. However, the heat shield felt metal material did abrade indicating a minimum clearance was achieved without damaging the turbine rotor or abradable heat shield shown in Figure 19. A new abradable heat shield was installed in the turbocharger and evaluated on the engine. Again, no measurable difference was seen in brake specific fuel consumption indicating no turbine efficiency improvement. Engine testing was also completed with an abradable power turbine heat shield, with no measurable performance benefit.

The abradable heat shield did not improve turbine efficiency because windage losses due to viscous shear are negligible below .020"-.030" backface clearance. However, there is a potential improvement in mechanical reliability, since insufficient backface clearance without abradable material would cause metal-to-metal contact, damaging both the turbine rotor and heat shield.

#### 5.0 Advanced Engine Dynamometer Testing: Task IV

The advanced turbocompound engine was equipped with the following hardware, which differed from the interim turbocompound configuration, to measure the cumulative performance gains of the advanced engine:

- . Redesigned 1.8 inch diameter pulse exhaust manifold.
- . New injector camshaft lobe with improved injection characteristics.
- . Insulated exhaust manifold, charge air turbine volute, and interstage duct.
- . Simulated air-to-air aftercooling ( $110^{\circ}$  IMT).
- . Simulated ball bearing system for the power turbine.
- . Optimum power turbine gear ratio.
- . Abradable compressor shroud with .004 inch axial clearance.

Emissions were measured using the 13-mode gaseous emission cycle to verify that the engine was operating at the combined 6 gram level. A 13-mode BSNO<sub>x</sub> of 5.66 gm/bhp-hr and BSHC of .27 gm/bhp-hr was measured at  $14^{\circ}$  BTDC dynamic injection timing with advanced injection timing to  $21^{\circ}$  BTDC in the light load modes.

A steady-state dynamometer performance map was generated with the advanced turbocompound engine at the 6 gram combined emission

Figure 19

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ABRADABLE HEAT SHIELD

level. Figure 20 shows the isofuel consumption islands as a function of engine speed and power. Brake specific fuel consumption at rated power was .310 lb/bhp-hr. The minimum brake specific fuel consumption of .298 lb/bhp-hr occurred at 1500 rpm, which is equivalent to a thermal efficiency of 46.4 percent. Fuel consumption at torque peak power was .300 lb/bhp-hr. A comparison of fuel consumption along the torque curve for the advanced versus interim turbocompound engine is shown in Figure 21.

#### 6.0 VMS Analysis: Task V

The advanced turbocompound performance map was input into Cummins' VMS program to predict the tank mileage over the Cummins' Pilot Center fuel economy route for comparison to the VMS results of the interim turbocompound engine. A summary of the VMS calculations presented in Table 1 shows a predicted tank mileage of 5.75 mpg for the advanced turbocompound engine while the interim turbocompound engine prediction was 5.40 mpg. Thus, a VMS predicted tank mileage improvement of 6.5% was achieved. This produced a fuel savings of 2.9 gallons over the simulated 260.17 mile course.

A VMS comparison was also made for the interim turbocompound field test route from Florida to California. The southern route across the United States includes sections of I-75, I-10, I-20, and I-8. This route, shown in Figure 22, provides a variety of terrains including plains, rolling hills, and mountainous grades. The VMS predictions are shown in Table 2. The advanced engine achieved tank mileage improvements of 6.9 and 6.1 percent over the interim engine at maximum cruise speeds of 60 mph and 65 mph, respectively. A VMS data summary for five types of terrains is presented in Table 3 for the advanced turbocompound engine and interim turbocompound engine. VMS predicted tank mileage improvements for the advanced turbocompound engine range from 7.0 percent on level terrain to 6.2 percent on a mountainous route.

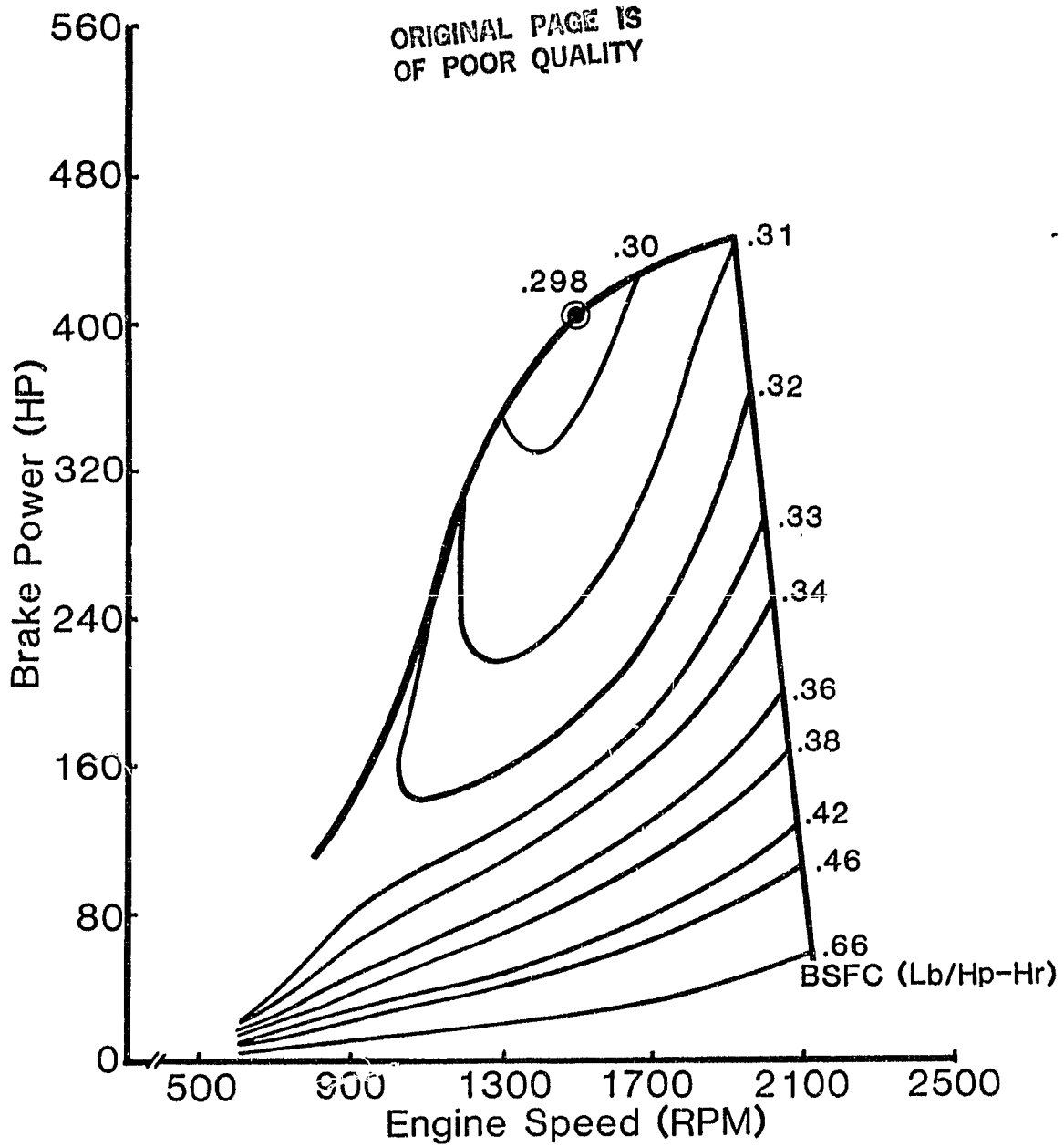
#### 7.0 Discussion of Results

The advanced turbocompound engine development was pursued by Cummins and DOE with the objective of a 5% improvement in fuel economy over the vehicle test (interim) turbocompound engine (Ref. 1). Engine improvements could be realized by increasing the available energy in the exhaust gas at the turbine inlet, by reducing operating clearances with abradable shrouds in the turbo-machinery, and through refined system optimization.

The available energy in the exhaust gas at the turbine inlet was increased through external insulation of the exhaust system components and by a new exhaust manifold design. The exhaust system insulation increased the turbine inlet temperatures 10<sup>0</sup>-15<sup>0</sup>F resulting in an .002 lb/bhp-hr torque curve BSFC improvement. The exhaust manifold design improved the pulse conservation of the exhaust blowdown. This improved torque curve BSFC from .003 lb/bhp-hr at 1900 rpm to .005 lb/bhp-hr at 1300 rpm.

Power turbine shaft mechanical losses were reduced by using ball bearings versus the interim journal bearing design. The power

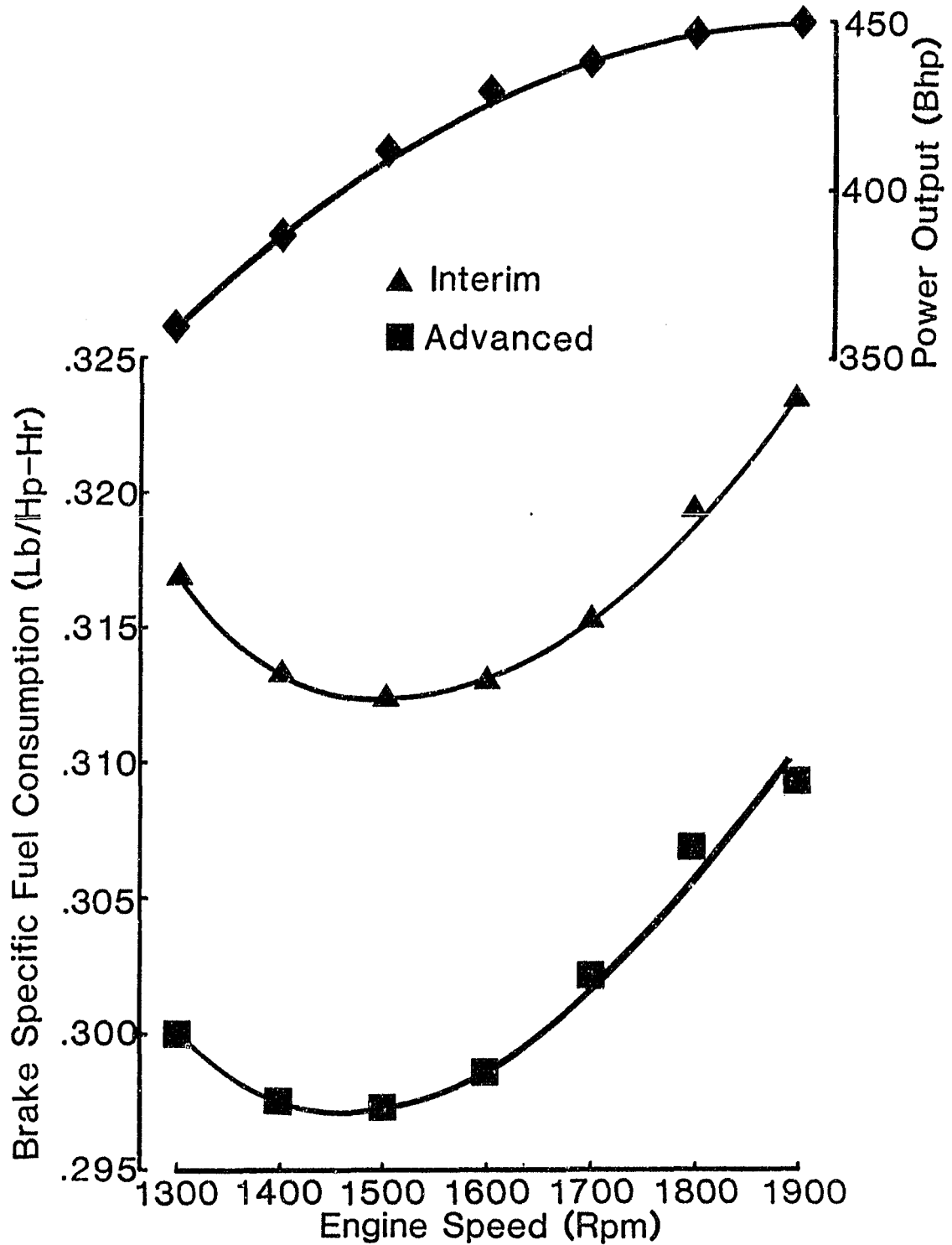
Figure 20



PERFORMANCE CURVE  
ADVANCED TCPD-450

Figure 21

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INTERIM AND ADVANCED TCPD  
ENGINE PERFORMANCE

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Table I  
VMS DATA SUMMARY FOR  
CUMMINS FUEL ECONOMY ROUTE  
ADVANCED TCPD-450 VS. INTERIM TCPD-450

ENGINE	INTERIM	ADVANCED
MAXIMUM CRUISE SPEED (MPH)	55	55
AVERAGE VEHICLE SPEED (MPH)	43.3	43.3
FUEL USED (GALLONS)	48.2	45.3
AVERAGE FUEL RATE (LB/HR)	57.0	53.5
TANK MILEAGE (MPG)	5.40	5.75
% ADVANCED TCPD-450 MILEAGE IMPROVEMENT		6.5

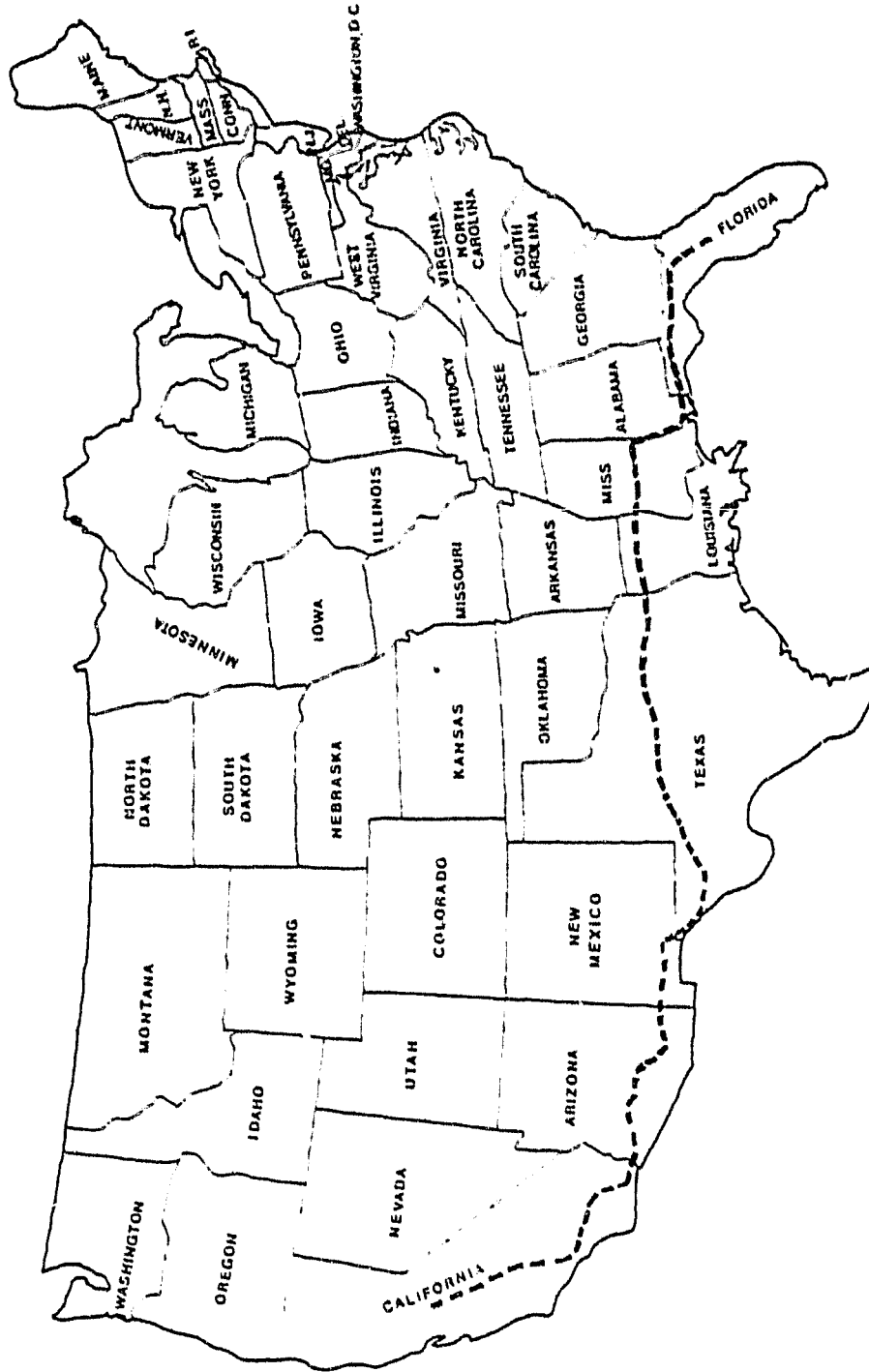
VEHICLE TEST INPUT CONDITIONS:

GW (LB)	73,000
CRUISE SPEED (MPH)	55
WIND SPEED (MPH)	9
WIND DIRECTION (DEG.)	219
TEMPERATURE (DEG. F.)	51
TRUCK: PILOT CENTER UNIT 30	
KENWORTH CONVENTIONAL	

Figure 22

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TURBOCOMPOUND FIELD TEST ROUTE



OPERATOR - Florida Refrigerated Services  
GVW- 72,000 Lbs.  
FREIGHT - Fruit & Produce

--- Test Route



Table II  
VMS DATA SUMMARY  
TCPD-450 FIELD TEST ROUTE  
ADVANCED TCPD-450 VS. INTERIM TCPD-450

Engine	Interim	Advanced	Interim	Advanced
MAXIMUM CRUISE SPEED (MPH)	60	60	65	65
AVERAGE VEHICLE SPEED (MPH)	57.0	57.1	60.6	60.6
FUEL USED (GALLONS)	470	440	503	474
AVERAGE FUEL RATE (LB/HR)	69.9	65.5	79.5	74.9
TANK MILEAGE (MPG)	5.79	6.19	5.41	5.74
% ADVANCED TCPD-450 MILEAGE IMPROVEMENT		6.9		6.1

#### TRUCK SPECIFICATIONS

TCPD-450 GEARED SPEED = 74.4 @ 1900 RPM - 19.0 HP FAN

INPUT CONDITIONS: GW = 73,000 LB

RADIAL PLY TIRES

STILL AIR

AMBIENT TEMPERATURE = 85°F

KENWORTH CONVENTIONAL TRUCK

ACCESSORIES: AIR CONDITIONING AND LOCKED FAN

ROUTES SIMULATED FROM TAMPA, FLORIDA, TO

LOS ANGELES, CALIFORNIA

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Table III

VMS DATA SUMMARY FOR VARIOUS TERRAINS  
ADVANCED TCPD-450 VS. INTERIM TCPD-450

ROUTE	ENGINE	TANK MILEAGE	% ADVANCED TCPD TANK MILEAGE IMPROVEMENT
LEVEL INTERSTATE	INTERIM	6.17	
	ADVANCED	6.60	7.0
ROLLING PLAINS	INTERIM	6.20	
	ADVANCED	6.62	6.8
.2 PERCENT UP INTERSTATE	INTERIM	5.66	
	ADVANCED	6.05	6.9
HILLY INTERSTATE	INTERIM	5.99	
	ADVANCED	6.39	6.7
MOUNTAIN PASS	INTERIM	5.20	
	ADVANCED	5.52	6.2

INPUT CONDITIONS: 18 HP ACCESSORIES, GEARED SPEED = 67 MPH,  
RADIAL PLY TIRES, 73,000 LB GW, 85°F,  
STILL AIR

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turbine efficiency was improved via optimizing the gear ratio or power turbine speed to the engine operating characteristics. The combination of these power turbine improvements resulted in an average torque curve BSFC improvement of .0015 lb/bhp-hr.

The fuel injection characteristics were improved with a new injector camshaft lobe which increased the injection pressure and shortened the injection duration compared to the baseline cam. The resultant increased combustion efficiency improved torque curve BSFC by .002-.003 lb/bhp-hr.

The interim turbocompound intake manifold temperatures were reduced 30°F to 110°F to reduce the nitric oxide emissions and increase the engine thermal efficiency. Combustion efficiency was improved with the resultant higher air/fuel ratios and lower nitric oxide emissions with lower combustion temperatures. A torque curve BSFC improvement of .001-.004 lb/bhp-hr was measured.

Abradable shrouds were applied to both the charge air turbine volute and compressor housing. Tip clearance losses are a function of a dimensionless clearance ratio:  $t/b-t$  where  $t$  = shroud to rotor clearance and  $b$  = total flow passage width. For the compressor, the clearance ratio was reduced at the outlet from approximately .100 to .02. A 1-2% compressor efficiency improvement was measured on both the bench test and engine performance test. BSFC was reduced .001 to .003 lb/bhp-hr along the torque curve.

The charge air turbine clearance ratio was reduced at the inlet from approximately .047 to .018. Both the engine performance test and bench test did not show turbine efficiency improvement. The reduction in the turbine clearance ratio was very small compared to the compressor and did not have a significant measurable effect on efficiency. A more detailed turbomachinery research program would be required to better understand clearance effects on the scalloped radial in-flow turbine and the centrifugal compressor.

The cumulative performance gains were evaluated by inputting the advanced turbocompound fuel map into Cummins' VMS program. The VMS calculations predicted tank mileages of 5.75 versus 5.40 mpg for the advanced and interim turbocompound engines, over the Cummins Pilot Center fuel economy route or a 6.5% improvement. In comparison to the vehicle test NH-400 horsepower reference engine, the advanced turbocompound achieved a fuel consumption improvement of 21.6%.

As discussed in the text of this report, there are improvements made to the interim and advanced turbocompound engines which were intended to improve the engine's performance under turbocompound conditions, but they also improve an equivalent turbocharged engine. These design changes cloud the benefit due to turbocompounding alone unless the effects of these modifications can be sorted out from the comparison. This performance testing was not performed with the advanced turbocompound engine.

### CONCLUSION

The primary objective of the advanced turbocompound diesel engine program was to improve the tank mileage by 5% over the 1980 vehicle test (interim) turbocompound diesel engines. Engine improvements used to develop the advanced turbocompound engine were:

- I. Increased the available exhaust gas energy to the turbines with a more efficient exhaust manifold and by insulating the exhaust system components.
- II. Improved the fuel injection characteristics by providing higher injection pressures and shorter injection duration.
- III. Improved the turbocompound system by optimizing the power turbine speed for maximum turbine efficiency and by reducing the turbine shaft bearing mechanical losses.
- IV. Lowered the intake manifold temperature (from 140°F to 110°F) to reduce nitric oxide emissions, and increase engine thermal efficiency.
- V. Improved the compressor efficiency 1-2% by reducing the operating clearances with an abradable shroud.

The combined effect of these improvements resulted in a rated power BSFC of .310 lb/bhp-hr with a minimum BSFC of .298 lb/bhp-hr while meeting the California 6 gram (BSNO<sub>x</sub>+BSHC) gaseous emission level.

The advanced engine performance map was used as input to Cummins' Vehicle Mission Simulation (VMS) program to predict the tank mileage over the Cummins' Pilot Center fuel economy route for comparison to the interim engines. In the course of the vehicle testing program completed in 1980, it was proven that an excellent correlation exists between VMS predicted fuel consumption - and actual vehicle test results. The VMS calculations predicted a tank mileage of 5.75 mpg for the advanced turbocompound engine while the interim turbocompound engine prediction was 5.40 mpg. Thus, a predicted tank mileage improvement of 6.5% was achieved with the advanced turbocompound diesel engine.

The advanced turbocompound engine offers significant improvements in specific fuel consumption. There is a progressive improvement in fuel consumption as a function of engine load with the maximum benefit occurring along the engine's torque curve. As the heavy-duty automotive vehicles typically operate at high load factors, they are particularly suited to the performance gains available by means of turbocompounding.

As fuel costs continue to rise in the future in real dollars, the turbocompound device will become a cost effective means of recovering exhaust energy.

It should also be noted that while the turbocompound engine was demonstrating reduced fuel consumption, it was conforming to

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more stringent environmental emission standards established by the State of California in 1980. The technique commonly employed to achieve lower NO<sub>x</sub> emissions is to retard the combustion process. This results in a degradation of the engine's thermal efficiency and increases the energy content of the exhaust gases. The turbo-compounding system is better able to utilize this otherwise wasted thermal energy and thus maintains a higher thermal efficiency.

In summation, the advanced turbocompound diesel engine program met and exceeded all tank mileage goals, further enhancing the potential fuel consumption savings of the turbocompound diesel engine. The turbocompound engine provides an opportunity for the future by offering increased thermal efficiency, reduced exhaust emissions, and improved driveability while maintaining present standards of durability.

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9.0 APPENDIXES

A. METRIC CONVERSION TABLE

CONVERSION FACTORS FOR SI (METRIC) UNITS

<u>Quantity</u>	<u>Conversion</u>	<u>Factor</u>	
Length	in to m	2.540	E-02
	mi to km	1.609	E+00
Area	in <sup>2</sup> to m <sup>2</sup>	6.451	E-04
Volume	in <sup>3</sup> to m <sup>3</sup>	1.638	E-05
	gal to l	3.785	E+00
Velocity	mi/hr to km/hr	1.609	E+00
Torque	lbf-ft to N-M	1.356	E+00
Pressure	lbf/in <sup>2</sup> to Pa	6.895	E+03
Power	hp to w	7.457	E+02
Mass	lb to kg	4.536	E-01
Temperature	°f to °c	$t_c = (t_f - 32) / 1.8$	
Fuel Consumption	lb/bhp-hr to g/kwh	6.083	E+02
	mi/gal to km/l	4.251	E-01
Emissions	gm/bhp-hr to g/kwh	1.341	E+00

REFERENCES

1. Brands, Michael C. et al: Vehicle Testing of Cummins Turbocompound Diesel Engine. NASA CR-159840, June, 1980.